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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 41

FORTY-FIRST ANNUAL MEETING
BUFFALO, N. Y., JANUARY 28-30, 1935

SEMI-ANNUAL MEETING
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TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1000

FORTY-FIRST ANNUAL MEETING, 1935

THE 41st Annual Meeting of the A. S. H. V. E. was attended by 350 members and guests during January 28-30, at the Hotel Statler, Buffalo, N. Y. The technical sessions, at which committee reports and 12 technical papers were presented, were noteworthy for their interested and continued attendance. An enjoyable entertainment program had been arranged by the Committee on Arrangements of Western New York Chapter and the efficiency and enthusiasm of the local members in the performance of their duties were apparent.

A meeting of the Council took place on January 27 and registration began at 8:30 a. m. on January 28.

Pres. C. V. Haynes, Philadelphia, Pa., opened the 1935 Annual Meeting of the Society in the Terrace Room of the Hotel Statler and introduced L. A. Harding, Buffalo, N. Y., chairman of the Reception Committee. Mr. Harding extended greetings on behalf of the Western New York Chapter and a hearty and cordial welcome, to which President Haynes responded briefly.

The Report of the Council was read by the Secretary.

Report of the Council

Since the last Annual Meeting the Council has had four meetings, February 8, June 19, October 22 and January 27, and it has carried on the business affairs of the Society as required by the Constitution and By-Laws. At the organization meeting of the Council, Pres. C. V. Haynes announced the personnel of Council Committees, and Special Committees and Technical Committees were selected.

The Secretary, A. V. Hutchinson, was reappointed and the Budget prepared by the Finance Committee was adopted. This Budget providing for an income of $49,277.00 exclusive of Research and an expense of $49,100.00 has been closely adhered to on the expense side and the income has been materially increased because of improved

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dues collections and the extremely favorable showing of THE GUIDE as given in detail by the Reports of the Finance and Guide Publication Committees.

Because meetings of the Council have been less frequent during the past two years, much of the business was again handled by correspondence. This volume of correspondence was augmented by activity on the Engineers Code directed by Mr. John G. Eadie, the Society's representative. The Council approved recommendations for publication of THE GUIDE 1935, and established a budget for printing 8500 copies.

Depositories for Society funds were selected. Various invitations inviting the Society to hold future meetings in many parts of the country were received and discussed.

The invitation of the Illinois Chapter to have the 42nd Annual Meeting, January, 1936, in Chicago was accepted and arrangements have been made to hold the Heating and Ventilating Exposition in International Amphitheatre, Union Stock Yards.

As required by the By-Laws, the Council nominated five members to serve on the Committee on Research for a term of three years and appointed Prof. F. E. Giesecke as the Society's representative on the National Research Council.

At the Council's October meeting in Buffalo, approval was given to the Annual Meeting program and the dues rate for 1935 was announced. The request of the Illinois Chapter to establish a limited chapter membership was granted and approval of the new Constitution and By-Laws for the Massachusetts Chapter was given.

In accordance with the provisions of Article B-II, Section 1 of the By-Laws, resignations of members were accepted and the cancellation of membership as required by Article B-II, Section 2, totaled 377; and elections of 293 gave a net loss of only 84 this year. Every possible means was used to collect outstanding dues in arrears and a special effort was made throughout the year to have as many members as possible retain their affiliation but it was necessary to cancel many memberships on December 31st.

The Council is glad to report to the membership that the Society's financial affairs are in good condition and that there is an indication of growing interest in the Society's activities and a definite trend toward a larger membership for 1935.

Respectfully submitted,

THE COUNCIL.

On motion of J. D. Cassell, Philadelphia, Pa., which was seconded and carried, the Report of the Council was received and filed.

The Report of the Secretary was then introduced.

Report of the Secretary

This brief review of the work done by the headquarters office staff during the past 12 months will serve to indicate the many activities involved in conducting the Society's business. The year 1934 started with a smaller personnel than had been employed since 1920. To carry on the Society's work effectively necessitated a rearrangement of assignments in order to divide that work which could not be curtailed. As events transpired, however, the volume of business showed a substantial increase and routine duties were very heavy because of better dues payments, an increased demand for copies of THE GUIDE 1934, and an increase in GUIDE advertising of almost 40 per cent over the previous year.

Each month material was obtained and prepared for publication in the JOURNAL and after the papers were reviewed and approved by the Publication Committee, styling, editing and proof reading were handled by the office staff in addition to the preparation of news notes and the chapter reports furnished by the local secretaries. About 300 pages of JOURNAL material were presented in the columns of *Heating, Piping and Air Conditioning*.

Due to the increased size of THE GUIDE 1935, which will come to members about

February 25th, the work of styling, proof reading, making-up and checking this volume of over 1000 pages, presented a difficult problem so that additional temporary help had to be obtained during the last few months of the year. To give some conception of just one portion of THE GUIDE work, that of proof reading, it might be visualized by comparing it with reading 20 full sized novels and then remembering that all of this must be done within a period of a little over two weeks.

At the same time that the text was being prepared, another phase of THE GUIDE work was actually under way, namely that of securing contracts from manufacturers for catalog data space. Only limited personal contact was feasible, so that a great portion of the sales promotional effort had to be made by mail campaigns. This program was energetically carried on from September to January and a total of 15,000 letters was sent out.

Earlier in the year, from February to September, an active mail campaign was carried on to sell copies of THE 1934 GUIDE and by October 15 none of this edition of over 8000 copies could be supplied. The sales exceeded the budget estimate by about $3000. In this campaign, over 30,000 mailing pieces were sent out.

In addition to handling all of these individual orders, preparing labels for shipment and then billing and collecting several thousand accounts demands much time.

During the year 325 applications for membership were received and these were prepared for the Committee on Admission and Advancement. When an application comes in the minimum number of letters required before a candidate can be notified of his election is 8 provided it is not required to write for dues check, follow-up proposers and seconders or write the candidate for supplementary information.

An important function of the office is the matter of keeping mailing addresses up-to-date. On every dues bill sent to members there is a place provided for giving address changes and members are urged to cooperate in notifying the Secretary's office of any change in their business or location. Frequently this is neglected and mail is delayed in delivery or returned.

In the case of a copy of THE GUIDE or TRANSACTIONS this is very expensive and adds to the postage expense. A typical example would be shipment of THE GUIDE to Chicago, costing 24¢. If it is returned because of non-delivery the expense is another 24¢ and when we find out the new location and again make shipment at some later date there is another charge.

Because of the shifting activities of members during the past few years, changes on the mailing list have been exceedingly heavy and costly. To illustrate the condition existing, in 1934 the records show nearly 800 changes of addresses in addition to 293 newly elected members and the revisions required by the loss of members. It should be remembered that these changes must be recorded on four separate files and in the case of members affiliated with chapters, the local secretaries must be notified.

During the year, Annual and Semi-Annual Meetings have been organized and the headquarters office in cooperation with the Meetings Committee is responsible for the details of the technical programs. This involves securing committee reports, technical papers and speakers, cooperating with the local chapter officers and committees which handle the entertainment functions and local arrangements in the meeting city.

In 1934 the Society headquarters again functioned as a clearing house for engineers engaged in heating, ventilating and air conditioning work and kept them informed of developments on the Engineers' Code for the Construction Industry. John G. Eadie, consulting engineer of New York, acted as the Society's representative and served as a member of the Code group which submitted the revised Code for approval. Because of changes made by NRA divisions in the draft approved by the engineering group, the Code has remained dormant.

The effort of the Society to obtain employment for members has been continued actively during the year and many leads have been furnished and a considerable number of placements have been recorded with mutual benefit to employers and

those who have been looking for employment. This work entailed heavy correspondence during every month of the year.

Three new Codes were added to the list of A. S. H. V. E. Standards in 1934, as a result of a letter ballot vote by Society members. These Codes cover the subjects of Testing and Rating of Return Line Low Vacuum Heating Pumps, Testing and Rating Air Cleaning Devices Used in General Ventilation Work and Testing and Rating Concealed Gravity Type Radiation (Hot Water).

Several proposed changes in the Society's Constitution and By-Laws were voted on by members of the Society during the year and these related to the extension of time during which the Council is authorized to set the annual rate of dues and initiation fees, also the qualifications for student membership. The required notices and letter ballots were sent to members eligible to vote.

Both directly and through specially appointed members, there has been active cooperation with the local chapters and from time to time announcements and notices, list of subjects for discussion and available speakers have been furnished to local chapter officers.

In addition, there are numerous routine matters and correspondence to be handled, hundreds of inquiries from non-members requesting sources of information, reprints and codes, so that the headquarters office staff has a variety of duties to perform to carry on the work of the Society.

It is a pleasure to acknowledge the cooperation given to the Secretary by the Officers and Council Members, the Chapter Officers and Committees as well as the Research Laboratory, which assisted in handling some of the technical correspondence.

The members I feel sure will join the staff in congratulating one of its members, Miss Pauline Kurland, for her good fortune in surviving the Morro Castle disaster.

This report would not be complete without expressing appreciation to the regular members of the staff, the Misses Babcock, Converse, Kurland and Mildner, for not only the faithful performance of their regular duties but for their willingness to give extra time to the extent of more than 1000 extra hours during the year because of their interest and loyalty to the Society, without thought of additional compensation.

It is significant that when the cost of administration of Society activities is carefully analyzed and checked with the reports of comparable organizations, it is found to be exceedingly low.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

The Report of the Committee on Admission and Advancement was read by R. H. Carpenter, New York, N. Y., at the request of A. J. Offner, chairman of the Committee, who could not attend because of a broken leg.

Report of Committee on Admission and Advancement

During the calendar year 1934, the Committee on Admission and Advancement received and reviewed 231 applications for membership, which resulted in the election of 84 Members, 66 Associates, 33 Juniors, and 48 Student Members, or a total of 231 new members for the Society.

Your Committee wishes to impress upon members who act as proposers and seconders of candidates for membership the importance of familiarizing themselves with and explaining to the applicant the Constitutional requirements for the different grades of membership, and also to point out to them the necessity of giving fully detailed information as to their training and experience. Furthermore, the sponsors must give such confidential information necessary for the guidance of the Committee in determining the grade of membership to which the applicant is entitled. The lack

of such full information and cooperation was evident in many applications, and resulted in a few cases in great disappointment to the applicant in the grade of membership voted him. While the Committee in no case has tried to keep an applicant from the grade of membership he is entitled to, it could not and would not vote a higher grade than the Constitution and By-Laws permitted and the submitted information seemed to indicate. Membership in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS must not be too easily acquired, nor be based on insufficient experience or training. Membership must be of such high standing as to be of pride to the member and of respect to non-members.

Your Committee wishes to call attention to the great increase in the number of student members admitted during the past year, especially as this grade of membership is carried at a financial loss to the Society. The number of student members increased during the year from 125 to 173 members. If this rate of increase is maintained, the strain on the treasury may become of grave concern. Either the number of publications these members receive should be reduced, or the annual fee increased to cover at least the actual cost. Furthermore, both the applicant's age and number of years this grade of membership may be held should be strictly limited. We of the Committee do not think the proposed constitutional amendment will fully accomplish the desired results.

In conclusion, the members of the Committee wish to express their sincerest thanks and appreciation for the constructive help given them by proposers and seconders, Council Members, the National Secretary, and the office staff at Society's headquarters.

Respectfully submitted,

ALFRED J. OFFNER, *Chairman,*
JOHN G. EADIE,
E. NUTE SANBERN,

COMMITTEE ON ADMISSION AND ADVANCEMENT.

It was voted upon motion, duly seconded, that the report be accepted as read.

Mr. Carpenter gave the Report of the Finance Committee, of which he was chairman.

Report of Finance Committee

The presentation of the Finance Committee's Report is a much pleasanter task than it was a year ago. At that time a deficit was in sight—a deficit which was held at a low figure only because, in the middle of the year, the budget had been revised and some drastic reductions made. Mr. Howatt very wisely bore this situation in mind in preparing the budget for 1934 and, as a result, income was estimated as pessimistically as possible and expenses held to a figure that provided for the bare existence of the Society but little else. The extent to which this kind of planning was justified is best summarized by the simple statement that the 1933 deficit of $736.00 has been followed by a 1934 surplus of $8191.00—a fact that is a source of much gratification to the members of the Finance Committee. The results accomplished are, in a large measure, due to the hard work and cooperation of Secretary Hutchinson and his staff in the collection of dues and the careful watching of office expenses.

The income of the Society comes from three principal sources—Dues from Members, Editorial Contract and Guide Advertising and Sales. Dues collections from members for the calendar year 1934 have been $2778.00 in excess of the budget estimate. Guide advertising has been $5522.00 in excess and Guide sales $2849.00 in excess of budget expectations. The total gross revenue in excess of the budget estimate is $13,065.97, reduced to a net surplus of $8191.34 by the excess cost of the Guide over that anticipated due to the increased number of copies printed, fixed charges which were unbudgeted, accounts receivable, depreciation, etc. The accounts

receivable from members in arrears of dues amount to $9723.00 for the year 1934, with an additional $16,695.00 due for 1933 and prior years. Approximately $6000.00 of these sums are in the form of notes. In preparing the 1935 Budget, all unpaid dues for years prior to 1934 have been considered as valueless; for which reason only last year's arrears have been taken into consideration for budget purposes. Even this figure has been written down 75 per cent, with the result that the anticipated revenue from all unpaid dues is set at $2430.74.

During the year the Council, upon the recommendation of your Finance Committee, sold $5000 par-value bonds of the Metropolitan-Edison Co. Although these bonds had been listed at far below par, their market value, at the time of sale, was much more favorable, and the bonds were sold for $4731.65. This money is now on deposit awaiting investment in United States Government securities, as required by the By-Laws. Securities owned by the Society, exclusive of Research Funds, as of Dec. 31, 1934, cost $26,722.71, with a market value substantially higher than was the case a year ago.

The Society's capital account, of net worth, on Jan. 1, 1934, was $32,458.10. The corresponding figure as of Jan. 1, 1935, is $41,393.53, an increase of $8935.43. This increase is due to the operating profit as outlined previously, supplemented by inventory items and modified by the loss on the sale of the Metropolitan-Edison bonds.

The balance sheet, as of Jan. 1st of this year, shows the following figures. For purposes of comparison, the corresponding figures of a year ago are given also:

	1934	1933
Cash on hand	$26,074.96	$11,939.99
Collectible accounts	23,197.30	21,776.90
Total invested assets, at cost	28,740.60	30,721.38
Other assets and inventories	5,222.46	4,866.79
Notes receivable	653.50	455.36
Total current assets	53,412.21	36,781.48
Total assets	84,498.13	69,760.42
Current liabilities	17,965.12	15,372.77

From the above it will be seen that the current assets are 2.97 times the current liabilities, from which we may conclude that the Society is in a sound financial condition.

Respectfully submitted,

R. H. CARPENTER, *Chairman*,
F. C. MCINTOSH,
J. F. MCINTIRE,

FINANCE COMMITTEE.

January 27, 1935.

Report of Certified Public Accountant

January 19, 1935

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
51 MADISON AVE.
NEW YORK, N. Y.

Gentlemen:

We have made an examination of your books and accounts for the Calendar Year ended December 31, 1934, and are submitting the attached Statements and Schedules as our report thereon.

In accordance with directions contained in the minutes of the meeting of the Society held in October, 1934, the Accounts Receivable arising from unpaid membership dues has been written down $12,105.20 due to members being dropped as follows:

Members	145
Associates	71
Juniors	20
Students	8
Total	244

The Reserve set up on the books against the remaining dues receivable, as suggested by Mr. A. V. Hutchinson, appear to be adequate to provide for future losses from this source.

The Accounts Receivable arising from advertising sold and other sources have been carefully examined, and all bad and doubtful accounts have been eliminated and ample reserves set up against future losses.

Inventories of Transactions for the years 1926 and 1927, in the amounts of $131.87 and $198.90 respectively, have been written off and new inventories set up for the years 1931 and 1932; amounting to $851.43.

Inventories of Emblems, Certificate Frames, Postage, Printing and Supplies were furnished by the Office Staff and were accepted as submitted without verification.

The details of Invested Assets consisting of Free Funds and Restricted Funds are shown on Schedule "2."

It should be noted that the Budget Estimate on Guide Expenses for 1935 contemplated an edition of 8500 copies, while costs shown cover an edition of 10,000 copies.

The payment of the item appearing on the Balance Sheet as "Due Research Laboratory" is contingent upon the collection of the unpaid dues as of December 31, 1934.

If any further data are desired we shall be pleased to furnish such from working papers on file at our office.

Respectfully submitted,

WILLIAM A. MILLIGAN & COMPANY,

Certified Public Accountants.

STATEMENT "A"

BALANCE SHEET

As of December 31, 1934

ASSETS

CURRENT ASSETS			
Cash in Banks (as per Schedule "1")		$25,974.96	
Cash on Hand, New York Headquarters		100.00	$26,074.96
Accounts Receivable	*Total*	*Reserve for Bad and Doubtful Accounts*	*Collectible Accounts*
Membership Dues	$13,690.04	$11,259.30	$ 2,430.74
Advertisements	23,185.08	2,685.57	20,499.51
Other Accounts Receivable	534.09	267.04	267.05
TOTAL ACCOUNTS RECEIVABLE	$37,409.21		
DEDUCT: Reserve for Bad and Doubtful Accounts		$14,211.91	
COLLECTIBLE ACCOUNTS			$23,197.30
NOTES RECEIVABLE			653.50
INVENTORIES (At cost)			
1928–1932 Transactions		$ 2,724.49	
Emblems and Certificate Frames		22.25	
Postage, Printing and Supplies		739.71	3,486.45
TOTAL CURRENT ASSETS (Brought Forward)			$53,412.21

STATEMENT "A"
(Continued)

ASSETS (Continued)

(Forwarded)			$53,412.21
INVESTED ASSETS (as per Schedule "2")			
Free Funds		$ 9,977.08	
Restricted Funds—Endowments		18,762.98	
TOTAL INVESTED ASSETS (at cost)			28,740.06
FIXED ASSETS			
Furniture and Fixtures	$ 5,215.46		
DEDUCT—Reserve for Depreciation	3,779.45	$ 1,436.01	
Library		300.00	
TOTAL FIXED ASSETS			1,736.01
DEFERRED AND PREPAID CHARGES			
Expenses of Annual and Semi-Annual 1935 Meetings		$ 109.85	
Chapter Meeting Allowance 1935		500.00	609.85
			$84,498.13

LIABILITIES AND CAPITAL

CURRENT LIABILITIES		
Sundry Accounts Payable		$ 5,664.25
RESERVE FOR COMPLETION OF 1935 GUIDE	$ 9,000.00	
Year Book	535.00	
Mailing 1935 GUIDE (Controlled Distribution)	2,000.00	11,535.00
DUE RESEARCH LABORATORY (Payable if and when collected—See Comments)		765.87
TOTAL CURRENT LIABILITIES		$17,965.12
DEFERRED INCOME—PREPAID DUES		376.50
RESERVE FOR 1933 AND 1934 TRANSACTIONS		6,000.00
ENDOWMENTS		
Society	$17,666.97	
F. Paul Anderson Award	1,096.01	18,762.98
CAPITAL ACCOUNT, being net worth of Society, December 31, 1934 (See Statement "B")—(Exclusive of Research Laboratory)		41,393.53
		$84,498.13

STATEMENT "B"

COMPARATIVE STATEMENT OF INCOME AND PROFIT AND LOSS

For the Years Ended December 31, 1933 and December 31, 1934

YEAR 1933		INCOME			YEAR 1934
		MEMBERS' DUES			
	$33,372.00	Renewals—Members and Associates		$28,980.00	
		DEDUCT—			
	$ 6,507.00	Cancellations	$5,166.00		
	19,344.94	12,837.94 Provision for Unpaid Dues	2,328.76	7,494.76	
	$14,027.06			$21,485.24	
	5,610.82	DEDUCT—40% to Research Laboratory		8,594.10	
	$ 8,416.24			$12,891.14	
	1,000.00	Renewals—Junior Members (net)		1,065.00	
	120.00	Renewals—Student Members (net)		318.00	
$ 9,536.24		TOTAL MEMBERS' DUES FROM MEMBERS			$14,274.14
		NEW MEMBERS' DUES			
	$ 1,480.50	Members and Associates		$ 1,945.50	
	592.20	DEDUCT—40% to Research Laboratory		778.20	
$9,536.24	$888.30	(Brought Forward)		$ 1,167.30	$14,274.14

STATEMENT "B" (Continued)

YEAR 1933				YEAR 1934
$9,536.24	$ 888.30	(Forwarded)	$ 1,167.30	$14,274.14
	185.00	Junior Members	284.50	
	237.00	Student Members	144.00	
1,310.30		TOTAL NEW MEMBERS' DUES		$ 1,595.80
....		Initiation Fees		....
$10,846.54		TOTAL INCOME FROM MEMBERS' DUES		$15,869.94
		INCOME FROM EDITORIAL CONTRACT		
	$14,666.70	Income	$14,000.04	
10,967.86	3,698.84	DEDUCT—Members' Subscriptions	3,727.54	10,272.50

INCOME FROM MISCELLANEOUS SALES

COST	SALES PRICE		SALES PRICE	COST	
$315.41	$473.57	Reprints and Books	$ 920.59	$ 845.57	
47.09	70.41	Transactions	792.82	400.16	
157.71		Transactions written off (1926–1927)	....	330.77	
....	....	Codes	6.10	149.93	
54.17	39.15	Emblems and Certificate frames	141.50	90.29	
	$583.13	TOTAL SALES PRICE	$1,861.01		
$574.38		TOTAL COST		$1,816.72	
8.75		GROSS INCOME ON MISCELLANEOUS SALES			44.29
763.21		INCOME FROM INVESTMENTS			897.27
$22,586.36		GROSS INCOME			$27.084.00

GROSS INCOME FROM GUIDE

$15,213.50	GUIDE Advertisements 1935 Edition		$21,022.00	
	COSTS			
	$1,338.39	Paper Costs	$2,447.52	
	2,925.00	Printing and Binding Costs to Date	2,600.75	
	5,000.00	Estimated to Complete	9,000.00	
	389.26	Engraving and Art Work	342.01	
	1,500.00	Editorial Service and Clerical Salaries	1,208.08	
	850.00	Mailing Expense (Controlled Distribution)	2,000.00	
13,259.70	1,257.05	Sales Promotion Advertising 1935 GUIDE	1,651.73	19,250.09
$ 1,953.80	GROSS PROFIT 1935 GUIDE ADVERTISING			$ 1,771.91
$ 9,248.86	GUIDE COPY SALES 1934 EDITION			$11,349.09
	COSTS			
	$1,934.39	Mailing and Postage	$ 973.40	
3,558.74	1,624.35	Sales Promotion Copy Sales	1,507.29	2,480.69
$ 5,690.12	GROSS PROFIT 1934 GUIDE (Before Deducting Production Costs)			$ 8,868.40
$22,586.36	$7,643.92	(Brought Forward)		$10,640.31	$27,084.00

STATEMENT "B"
(Continued)

YEAR 1933				YEAR 1934
$22,586.36	$7,643.92	(Forwarded)	$10,640.31	$27,084.00
	428.59	ADDITIONAL INCOME 1934 GUIDE ADVERTISING	185.74	
	8,072.51	DEDUCT	10,826.05	
	1,460.94	Provision for Bad Debts	3,000.00	
$ 6,611.57		TOTAL GROSS PROFIT FROM GUIDES		$ 7,826.05
$29,197.93		TOTAL GROSS INCOME FROM ALL SOURCES		$34,910.05
		EXPENSES		
	$13,571.00	Salaries Paid Secretary and Office Staff (Schedule "4")	$11,375.00	
	652.57	Traveling Expense—Secretary	793.41	
	629.12	Traveling Expense—President	700.88	
	154.41	Emergency Fund—President	113.25	
	2,676.45	Rent and Light	2,549.97	
	55.00	Moving Expense	....	
	448.94	General and Office Expense	468.06	
	342.55	Stationery and Supplies	320.77	
	538.97	Telephone Expense	517.75	
	171.01	Telegraph Expense	178.17	
	347.28	Multigraphing	444.78	
	521.55	Depreciation on Furniture and Fixtures	521.55	
	1,800.00	Reduction in H. P. & A. C. Contract	1,200.00	
	200.00	Professional Services	200.00	
	50.00	Contribution to Amer. Stand. Assn	50.00	
	192.81	Constitution and By-Laws Printing Expense	....	
	443.62	General Printing Expense	493.85	
	3,000.00	Allowance 1934 Transactions	3,000.00	
	1,562.48	Postage	1,595.02	
	510.40	Year Book	535.00	
	807.67	Meetings	971.72	
	1,000.00	Chapter Allowance	463.73	
	50.00	Filing System	....	
	50.00	Awards and Medals	50.00	
	158.33	Bank Charges	70.42	
	.....	Carpet	98.39	
$29,934.16	.....	Register of Copyrights	6.99	$26,718.71
$736 23		PROFIT OR LOSS FOR YEAR (STATEMENT "C")		$ 8,191.34
		CAPITAL ACCOUNT OF SOCIETY, January 1, 1934	$32,458.10	
		ADD—Transaction Inventory 1931–1932	851.43	
			$33,309.53	
		DEDUCT Loss on Sale of Metropolitan Edison Bonds	107.34	33,202.19
		CAPITAL ACCOUNT OF SOCIETY, DECEMBER 31, 1934 (See Statement "A")		$41,393.53

STATEMENT "D"

RESEARCH LABORATORY

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

BALANCE SHEET

As of December 31, 1934

ASSETS

CASH		
Bankers Trust, 42nd Street & 5th Ave., New York City	$ 190.16	
Forbes National Bank, Pittsburgh, Pa.	2,650.55	
On Hand, Pittsburgh, Pa.	50.00	
Bank for Savings, New York City	2,838.99	$5,729.70
DUE FROM SOCIETY		765.87
RESTRICTED FUNDS—Endowment		
Cash in Bank for Savings, 280 Fourth Avenue, New York City	$ 402.91	
Cash in Bank of United States, New York City (representing 40% of total deposits, 60% dividend having been paid in previous years	243.93	646.84
		$7,142.41

LIABILITIES

ENDOWMENT FUND	$ 646.84
CAPITAL ACCOUNT, being net worth of Research Laboratory, December 31, 1934 (Statement "E")	6,495.57
	$7,142.41

STATEMENT "E"

RESEARCH LABORATORY

COMPARATIVE STATEMENT OF INCOME AND PROFIT AND LOSS

For the Years Ended December 31, 1933 and December 31, 1934

YEAR 1933				YEAR 1934
		INCOME		
	$5,610.82	Share of Dues from A.S.H.V.E. Renewals	$7,501.53	
6,203.02	592.20	New Members	927.16	$ 8,428.69
		Receipts from Contributions (as per Schedule "5")		
	$ 625.00	Special	$4,345.31	
4,154.00	3,529.00	General	4,472.00	$ 8,817.31
135.00		Interest on Securities Owned		90.00
8.25		Interest on Bank Balances		.....
108.14		Refund—American Society Testing Materials Exhibit		.....
.....		Exposition (Net)		.....
.....		Interest on Endowment Fund		575.92
$10,608.41		GROSS INCOME		$17,911.92
		EXPENSES		
		Committee on Research Expense		
	$ 401.92	Traveling Committee and Laboratory Personnel	$ 819.96	
	500.00	Salary—Technical Adviser	2,000.00	
	70.30	General Printing and Promotion Expense	324.13	
	500.00	Correlating Thermal Research Expense	500.00	
	$1,472.22	TOTAL EXPENSES—Committee on Research	$3,644.09	
$10,608.41	$1,472.22	(Brought Forward)	$3,644.09	$17,911.92

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STATEMENT "E" (Continued)

$10,608.41	$1,472.22	(Forwarded)	$3,644.09	$17,911.92
		Laboratory Expense (Pittsburgh)		
	$8,491.70	Salaries (as per Schedule "6")	$6,869.95	
	472.54	Laboratory Supplies and Equipment	481.57	
	344.63	Office Supplies and Expense	261.23	
	.....	Meetings	304.31	
	$9,308.87	TOTAL LABORATORY EXPENSES (Pittsburgh)	$7,917.06	
		Cooperative Research Expense		
	$3,225.00	University of Illinois	$2,555.00	
	875.00	University of Minnesota	500.00	
	1,153.02	Harvard University	1,400.00	
	200.00	Yale University	.....	
	50.00	Miscellaneous	45.00	
	$5,503.02	TOTAL COOPERATIVE RESEARCH EXPENSES	$4,500.00	
		CONTINGENCY FUND EXPENSES		
	$ 259.81	Exposition Exhibit Expense	$ 85.00	
$16,543.92		TOTAL EXPENSES		$16,146.15
$ 5,935.51		PROFIT OR LOSS FOR YEAR (Statement "F")		$ 1,765.77
		CAPITAL ACCOUNT JANUARY 1, 1934	$4,898.31	
		DEDUCT—Loss on Sale of Securities	168.51	4,729.80
		CAPITAL ACCOUNT, DECEMBER 31, 1934		$ 6,495.57

Mr. Cassell moved that the report be accepted and filed and this was properly seconded and carried.

President Haynes introduced J. S. Snyder, Buffalo, N. Y., who reported for the tellers who recorded the vote on the amendment to the Constitution.

Report of Tellers

The votes cast for Amendments to the Constitution have been tabulated by the Board of Tellers and the following results are reported:

Art. C-II Sec. 7 ..... yes 388; no 9
Art. C-III Sec. 4 ..... yes 390; no 7
Total number of votes cast 447
Total legal votes 397

Respectfully submitted,

H. A. HEXAMER, *Chairman*,
WM. ROEBUCK, JR.,
C. H. LOVE.

January 26, 1935.

J. J. Landers, Buffalo, N. Y., gave the Report of the Tellers for Election of Officers as follows:

Report of Board of Tellers

The votes cast for Officers have been tabulated by your Board of Tellers with the following results:

President—JOHN HOWATT..... 416
1st Vice-President—G. L. LARSON..... 415
2nd Vice-President—D. S. BOYDEN..... 416
Treasurer—A. J. OFFNER..... 416

Members of the Council—3-year term:

J. F. MCINTIRE..... 414
ALBERT BUENGER..... 416
F. E. GIESECKE..... 416
W. E. STARK..... 416

The votes cast for Members of the *Committee on Research* have been tabulated by the Board of Tellers with following results:

Three-year term:

C. A. DUNHAM..... 415
W. L. FLEISHER..... 416
ELLIOTT HARRINGTON..... 416
A. P. KRATZ..... 416
H. C. MURPHY..... 416

Total number of votes cast 475
Total legal votes 416

Respectfully submitted,

L. A. CHERRY, *Chairman.*
R. T. THORNTON,
J. J. LANDERS.

January 26, 1935.

The Report of the Membership Committee was read by F. C. McIntosh, Pittsburgh, Pa., Chairman of the Committee.

Report of the Membership Committee

Since the last Annual Meeting, 293 members have been elected to the various grades of Membership, as follows:

Members	99
Associates	73
Juniors	37
Students	84
	293

There are about 55 applications under consideration.

Although no drive was undertaken, an attempt was made to present the advantages of membership to those who might be eligible.

The unusual feature this year has been the high percentage of students. The new sources of such members are:

Case School of Applied Science, Cleveland, Ohio.

University of Oklahoma, Norman, Okla.

University of Minnesota, Minneapolis, Minn.

It is hoped that expansion in this direction will be of distinct benefit to the younger men about to enter our field, and also to our Society, in that these men are likely to carry their affiliation beyond the student grade.

Progress has been made toward the establishment of chapters in the following cities:

XUM

Baltimore	Oklahoma City
Dallas	Omaha
Denver	San Francisco

Some or all of these groups, but particularly Baltimore, Denver and Oklahoma City, with help from us, can be in position to apply for charters within the coming year.

Present membership or geographical location suggests the formation of chapters in the following other cities:

Atlanta	New Orleans
Indianapolis	Vancouver
Montreal	Washington
	Winnipeg

And there are still other cities or districts with probably enough good men to make up an active chapter group. The Albany-Schenectady-Troy district is mentioned as an example.

As the trend of Society thought indicates an increasing interest in chapter activities on the part of the national organization, there is, or should be, a greater incentive for the establishment of additional chapters.

It is the recommendation of this committee that all of us attend and extend the activities of our local chapters where they exist, and where they do not, that we develop informal groups, if possible, to take their place until they can be formed.

We acknowledge with thanks the great assistance we have received from so many of our members that we are unable to name them here.

Respectfully submitted,

F. C. McINTOSH, *Chairman*,
ALBERT BUENGER,
W. A. RUSSELL,

MEMBERSHIP COMMITTEE.

On motion of Mr. Cassell, seconded and carried, the report was accepted and filed.

President Haynes presided at the Second Session and introduced John Howatt, Chicago, Ill., chairman of the Committee on Research, who gave the report.

Report of the Committee on Research

The Report of the Committee on Research was prepared by John Howatt, Chairman, and F. C. Houghten, Director of the A. S. H. V. E. Research Laboratory.

The Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held three meetings during the year, each attended by a large percentage of the members. At the organization meeting in February Mr. John Howatt, Chicago, Ill., was elected to serve as Chairman. A budget for the year was adopted and other items of business pertaining to adjustment of finances were transacted. A program of research for the year was also discussed and approved. Another meeting of the Committee was held in June during the Semi-Annual Meeting of the Society at Buck Hill Falls, Pa., when the research program was further developed and the activities of the various Technical Advisory Committees were discussed. At this meeting it was planned to publish a new booklet dealing with the Society's research activities and its benefit to industry. A committee under the chairmanship of J. H. Walker undertook this work, and an attractive booklet, Research—The Fact Finder, has since been printed and distributed. Another meeting of the Committee as scheduled was held during the 1935 Annual Meeting at Buffalo in January.

The Committee experienced difficulties in developing a budget which could be

balanced with the reduced income. Great satisfaction has been felt, however, in the fact that for the first time during the past few years the budget adopted at the beginning of the year has not required further retrenchment. In fact, the Committee was able to meet a number of obligations incurred during former years and still close its books with a better cash balance at the close of this year than has been customary. The 1935 Committee will, therefore, begin the new year without any back log of debts and with a small but comfortable balance in the treasury.

In order to extend the research program possible under the curtailed income, the Committee conceived and put into operation a plan of accepting donations from interested branches of industry to be applied specifically to certain self-liquidating research projects. In this connection a program of research was outlined and undertaken for the *Portland Cement Association* to determine the heat transfer coefficients for different types of concrete wall construction. Eighteen hundred dollars were pledged for this work, which is being carried on at the University of Minnesota in cooperation with the Committee. Plans for a research program were developed dealing with the problem of corrosion of copper and copper bearing alloys used in the heating industry, and the various branches of the industry were invited to participate; however, this plan has not materialized to date.

Besides carrying on a comprehensive research program, the Committee was able to cooperate in other Society work by furnishing some text material for THE GUIDE, 1935, in releasing the services of J. L. Blackshaw, a member of the Laboratory staff, to the Guide Publication Committee, and in handling a limited amount of technical correspondence received at the Headquarters Office of the Society.

A creditable exhibit of the activities of the Research Laboratory and the cooperating universities was prepared and shown at the Heating and Ventilating Show held at the time of the Annual Meeting in New York, and a less comprehensive exhibit was displayed at the annual convention of the *American Oil Burner Association* in Philadelphia, and some panels depicting the Research were on display at the Power Show in New York.

1934 Research Contributors

Aerofin Corp.
Allegheny County Steam Heating Co.
American Architect.
American Radiator and Standard Sanitary Corp.
Barber-Colman Co.
Buffalo Forge Co.
Bush Manufacturing Co.
Carrier Engineering Corp.
Crane Co.
Duquesne Light Co.
General Electric Co.
General Motors Co., Frigidaire Div.
Grant Bldg., Inc.
Hoffman Specialty Co.
Ilg Electric Ventilating Co.
International Exposition Co.
International Union of Operating Engineers.
Johns-Manville Corp.
Johnson Service Co.
Kewanee Boiler Co.
Kinetic Chemicals Co.
Mine Safety Appliances Co.
Modern Building Insulation Co.
Minneapolis-Honeywell Regulator Co.
Mueller Brass Co.
Nash Engineering Co.
National Regulator Co.
National Warm Air Heating and Air Conditioning Assn.
Herman Nelson Co.
J. J. Nesbitt, Inc.
New York Blower Co.
Pittsburgh Meter Co.
Portland Cement Assn.
B. F. Sturtevant Co.
Trane Co.
United Engineers and Constructors, Inc.
Utilities Research Commission, Inc.
Westinghouse Elec. & Mfg. Co.
York Ice Machinery Corp.
Young Radiator Co.

Research Activities During 1934

During the year 1934 the Committee on Research gave consideration to 16 research projects, each of which was considered by a Technical Advisory Committee. In some cases the work of the committees resulted in improvement or standardization of available data, while in other cases the committees developed a program of research to be carried out whenever the necessary funds and facilities are available. Because of the budget limitations, both for the work at Pittsburgh and in cooperating Universities, actual investigation was limited to a few of these projects. In some instances only a small amount of time could be given to a subject, with the result that solution of the problems considered by the Committee on Research has progressed slowly. Nevertheless, the Technical Advisory Committees were of great service in cooperating with the Chairman of the Committee on Research and the Director of the Laboratory in developing plans for carrying on the studies made. The organization of the various Technical Advisory Committees brought the combined experience and abilities of 94 engineers, educators and scientists into cooperation for the advancement of fundamental knowledge. Dr. A. C. Willard, president of the University of Illinois, continued as Technical Adviser to the Committee.

Cooperating Universities

The practice of extending the work carried on under the Committee's direction by cooperation with engineering departments of universities was continued. It is particularly gratifying to note that while the Committee has had to cease contributing financially toward many of these studies, the interest of the institutions has been retained, and in almost every instance active work has been continued. The following institutions have had active cooperative agreements with the Committee on Research during the year, under which definite projects have been investigated:

1. UNIVERSITY OF MINNESOTA: (1) Dust and dust control apparatus. (2) Heat transmission.
2. UNIVERSITY OF ILLINOIS: (1) Direct and indirect radiation with gravity air circulation. (2) Cooling of buildings in summer.
3. HARVARD SCHOOL OF PUBLIC HEALTH: (1) Minimum air requirements for ventilation.
4. UNIVERSITY OF WISCONSIN: (1) Heating of buildings.
5. AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS: (1) Insulation of buildings.
6. MICHIGAN COLLEGE OF MINING AND TECHNOLOGY: (1) Corrosion in return lines in relation to the chemical composition of the water, vapor, and gas handled.
7. CASE SCHOOL OF APPLIED SCIENCE: (1) Heat transfer from direct and extended surfaces with forced air circulation.
8. YALE UNIVERSITY: (1) Oil burning equipment used in heating.
9. MARQUETTE UNIVERSITY: (1) Methods of air supply and distribution in air conditioning.

Papers Resulting from 1934 Research

The Committee on Research, through its Technical Advisory Committees, succeeded in bringing its studies to a sufficient stage of completion to result in 14 technical papers for presentation before the Society and publication[1] in the 1934 and 1935 issues of the A. S. H. V. E. JOURNAL (in *Heating, Piping and Air Conditioning*), as follows:

1. Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, March, 1934.
2. Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, by L. E. Seeley and E. J. Tavanlar, May, 1934.
3. Insulating Value of Bright Metallic Surfaces, by F. B. Rowley, June, 1934.
4. Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, by G. L. Tuve and C. A. McKeeman, June, 1934.
5. Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick; July, 1934.
6. Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin; August, 1934.
7. Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet; September, 1934.

[1] These papers also appear in A. S. H. V. E. TRANSACTIONS: 1-7 inclusive in Vol. 40, 1934, and 8-14 inclusive in Vol. 41, 1935.

8. Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James; November, 1934.
9. Study of Unit Room Coolers in the Research Residence, by A. P. Kratz, M. K. Fahnestock and S. Konzo; November, 1934.
10. The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle; December, 1934.
11. Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten and Carl Gutberlet. Presented at the Annual Meeting, January, 1935.
12. A Laboratory Study of Minimum Ventilation Requirements; Ventilation Box Experiments, by W. H. Lehmberg, A. D. Brandt, and Kenneth Morse; January, 1935.
13. Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick; January, 1935.
14. A Carbon Monoxide Alarm and Ventilation Control, by F. C. Houghten and Linwood Thiessen; March, 1935.

Research Projects Investigated During 1934

The personnel of each of the 16 Technical Advisory Committees working with the Committee on Research are listed, with a report of their year's activity:

1. HEAT REQUIREMENTS OF BUILDINGS.—*Technical Advisory Committee:* D. S. Boyden, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.

The Committee on Heat Requirements of Buildings, appointed in November 1933, realized that it confronted a very difficult task, the solution of which has baffled heating engineers since the beginning of the profession. The committee proceeded to analyze current practices for estimating heating requirements of buildings and developed plans for attack on the problem from a number of points of view.

Analyses of Current Practices

The analyses of current practices indicated clearly that use of the more accurate heat transfer and infiltration coefficients developed by the Laboratory and cooperating universities during the past few years resulted in an unwarranted increase in installed heating capacity in modern buildings. This fact indicates a need of correcting assumptions regarding basic outside temperatures and wind velocities and their effect on heat loss from buildings. It also appeared desirable to further consider such factors as heat capacity of the building, sources of heat within the building, and solar radiation effects.

The analysis of the relative effect of outside temperatures and wind velocities on heat requirements made by the *Heating, Piping and Air Conditioning Contractors National Association* some years ago was considered. While the careful consideration given by them to the effect of concurrent low outside temperatures and wind velocities seemed entirely logical, their practice of allowing one degree fall in temperature as equivalent to one mile of wind velocity could not be accepted as a scientific solution of the problem.

An analysis of present GUIDE methods of calculating heating requirements indicates that too much consideration is given to the effect of wind velocities. THE GUIDE practice recommended at the time the Committee began its study based all calculations for all sides of a building on a 15-mph wind velocity and then added an additional 15 per cent for the heat loss from the two exposed sides. Consideration of this question has resulted in elimination of the 15 per cent exposure factor for exposed sides in the recommended practice given in THE 1934 GUIDE since published.

Research Carried on by the Committee

Prof. Axel Marin of the University of Michigan, in cooperation with the Detroit Edison Co. and the Committee, made a study of the heat loss from rooms at different

elevations in the Penobscot Building in Detroit, Mich., and presented a paper [2] on the subject at the Semi-Annual Meeting of the Society. This paper indicated a progressive decrease in heat loss with increased elevation in the building.

P. D. Close, Chicago, Ill., made an analysis of the relation between prevailing low temperatures and wind direction and velocities, resulting in the paper, Selecting Temperatures and Wind Velocities for Calculating Heat Losses, presented at the 1934 Annual Meeting of the Society. This paper adds much to the knowledge of probable combinations of low temperatures and high wind velocities resulting in maximum heat requirement.

The Research Laboratory at Pittsburgh collected data during the last heating season on the heating requirements of eight rooms at different elevations having different exposures in the Grant Building, a modern office building of the sky-scraper type in Pittsburgh. The results of this study were partially analyzed with a view of determining the relationships between wind velocities prevailing in close proximity to different exposures and elevations of the building to those reported by the Weather Bureau for the city. A paper [3] dealing with this phase of the subject was presented at the Semi-Annual Meeting of the Society, which indicates that the wind velocities against any particular exposure of the building cannot be predicted from Weather Bureau reports. Also, contrary to accepted practice in estimating heating requirements, the maximum velocities were not observed at high altitudes above surrounding buildings, but rather around the lower part of the building, where wind was confined between the buildings on the two sides of streets. The Laboratory is now proceeding with the analysis of the rest of the data collected last winter which should add something to our information on the relation of the required heating load to outside weather conditions.

J. H. Walker and G. H. Tuttle, Detroit, Mich., analyzed data on steam consumption in relation to types of buildings and occupancy, outside weather conditions and other factors, for 163 buildings in 22 different cities. The analysis was made as a contribution to the work of this committee resulting from data collected for the *National District Heating Association.* Quoting from the Summary of the paper [4] The Heat Requirements of Buildings, presented at the Annual Meeting of the Society:

"The paper gives the heat consumption of a large number of buildings on a degree-day basis and recommends, as a unit for comparison, the heat consumption per degree-day per thousand Btu of calculated heat loss.

"The maximum demands and load factors are given for different kinds of buildings. That the maximum demand and heat consumption are affected by other variables than temperature alone is demonstrated by typical load curves from a district heating system.

"Data on the hot water consumption of various kinds of buildings are included."

The Committee has plans for additional studies of the relation between low temperatures and wind velocities as reported by Weather Bureau stations for different cities, in order that better assumptions may be made for the worst prevailing conditions for heating in any locality.

The Committee has also outlined a study of the heating requirements of Sibley Dome, one of the University of Cornell Buildings in Ithaca, N. Y. This building was chosen for the reason that it was a modern university building with fairly constant daily occupancy. The following data will be collected during the coming heating season: a record of the rate of steam consumption; temperatures throughout the heating space; outside weather conditions, including temperature and wind velocity. The wide variations in temperature, together with the wide variations in wind velocity experienced during the heating season in Ithaca, suggest that this study should add much to knowledge of the variation in heat loss from a building with varying outside temperatures and wind velocities.

Studies of heat losses from buildings will be made at the University of Minnesota, University of Wisconsin and University of Illinois. The Research Laboratory is making a parallel study of cooling requirements of three offices in the Grant Building in Pittsburgh, a report [5] of which was presented at the Annual Meeting of the

[2] See No. 6 in List of Research Papers, p. 16.
[3] See No. 7 in List of Research Papers, p. 16.
[4] See No. 10 in List of Research Papers, p. 17.
[5] See No. 11 in List of Research Papers, p. 17.

Society. J. H. Walker has contributed a paper which will present a summary of the information which district heating companies have compiled on the load characteristics and annual steam requirements of various types of buildings, and will show steam consumption per degree day for various types of buildings, hourly demands, and water heating requirements.

Plans have been perfected for a study by Prof. F. B. Rowley of the University of Minnesota of variation in conductivity of concrete of various mix and aggregate.

Deductions from the Committee's Studies

The studies made during the year by the Committee indicate clearly that current methods for estimating heating requirements need revision, with probably the most serious errors in present methods resulting from improper consideration to the combined effect of concurrent low temperatures and high wind velocities. Basic outside conditions as pertaining to low temperatures and concurrent wind velocities must ultimately be based upon individual analysis for each city or locality similar to that made some years ago by the *Heating, Piping and Air Conditioning Contractors National Association,* and that suggested by Mr. Close in his recent paper. The Committee considers it too early, however, to base a final conclusion on the evidence at hand at this time. There is some evidence available resulting from the studies of the Research Laboratory and elsewhere which tends to indicate that wind velocities and directions reported by the Weather Bureau had little bearing on those prevailing in close proximity to buildings. If these indications are substantiated by further work, it is possible that a solution based upon Weather Bureau figures alone may be as much in error as present methods.

Other factors which appear to affect the heat loss from buildings are the heat capacity of the building, the elevation of individual rooms within the building and sun effects.

While the Committee feels that its studies to date do not warrant a final conclusive report solving the problems presented to it, it does feel that through analyses of present methods and research, marked progress has been made toward the solution, and that continued study of the subject along present lines will bring ideal solution of the problem progressively closer.

2. VENTILATION OF GARAGES AND BUS TERMINALS.—

Technical Advisory Committee: E. K. Campbell, *Chairman;* S. H. Downs, T. M. Dugan, E. C. Evans, F. H. Hecht, H. L. Moore, A. H. Sluss.

The objective of the Technical Advisory Committee over the past few years has been the development of a code for the ventilation of garages. In early attempts to prepare such a code the committee was confronted with the need for additional technical data particularly pertaining to elimination of carbon monoxide. Hence, the committee has included in its program research carried on by the Laboratory in Pittsburgh and by the University of Kansas in Kansas City and Lawrence, Kans., in cooperation with the Society.

The studies made at Pittsburgh and by the University of Kansas have been rather conclusive and demonstrate that in a ramp garage where there is considerable mixing of exhaust gases through the building, sufficient air must be supplied to dilute the exhaust gases to the point where the concentration of carbon monoxide contained therein is diluted to safe limits, or approximately one part of carbon monoxide in 10,000 parts of air. The studies made in single floor garages indicate that much can be accomplished by supplying outside air and exhausting vitiated air in such a manner as to eliminate the exhaust gases from the building before they have had an opportunity to mix with all of the air in the garage.

Because of the tremendous amount of air necessary for ventilating garages, a keen desire has been experienced for some instrument which could indicate and act as an alarm when the carbon monoxide concentration approaches a dangerous condition; or better yet, an instrument which can be used to control the operation of the ventilation equipment so as to keep the carbon monoxide concentration within safe limits. Largely through the need expressed by the work carried on under this com-

mittee and the encouragement given by the committee and the Laboratory in Pittsburgh, the Mine Safety Appliances Co. of Pittsburgh, holding the fundamental patents on which the Bureau of Mines carbon monoxide recorder is based, has been induced to develop an instrument suitable for use in garages. This instrument has been studied by the Laboratory in Pittsburgh in order to demonstrate its practicability in garage ventilation and with a view of adopting it as an instrument to control garage ventilation. As hoped for this study was completed in time for the presentation of a paper [6] on the study to the Society at the 1935 Annual Meeting.

The committee is continuing to work on the development of a code for the ventilation of garages.

3. SOUND IN RELATION TO HEATING AND VENTILATION.

—*Technical Advisory Committee:* V. O. Knudsen, *Chairman;* C. M. Ashley, C. A. Booth, F. C. McIntosh, R. F. Norris, J. S. Parkinson, C. H. Randolph, J. P. Reis, G. T. Stanton.

The committee has undertaken three projects which seem to be of most importance in relation to the acoustical problems which arise in the heating and ventilating of buildings.

1. J. S. Parkinson of Johns-Manville Co. is undertaking an investigation of the transmission of sound through ventilating ducts, with the specific purpose of determining quantitative data which can be used in calculating the effective sound reduction accomplished by different kinds of ducts, lined with different types of absorptive material, or including different types of sound filters.
2. C. M. Ashley of Carrier Engineering Corp. is undertaking an investigation to determine the noise rating of different types of machines used in the heating and ventilating of buildings.
3. Data are being collected on the amount of noise which can be tolerated without discomfort or objection, in different types of buildings. The acoustical engineers of the Electrical Research Products, Inc., have many men in the field equipped with facilities for making these measurements, and it is hoped that within a year or so, tolerance limits can be established for different buildings.

When the amount of noise which different machines generate is known, and when the amount of noise which can be tolerated in different kinds of buildings is also known, it is then possible to specify just how much sound insulation or sound absorption must be provided to meet any specified guarantee of noise.

The committee will welcome suggestions with regard to any of the above mentioned, or other, projects.

4. AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT.

—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* W. L. Fleisher, D. E. French, J. J. Aeberly, R. R. Sayers, C.-E. A. Winslow.

The cooperative study between the Society and Harvard School of Public Health on minimum air requirements for ventilation was continued during the year. An outline of this work appeared in the annual report of the committee for 1933, and the results of preliminary research have been presented [7] at the Annual Meeting. The work is scheduled for completion during next summer, and the final report will be presented at the following Annual Meeting.

Other suggestions considered by the Committee for future research are numerous. However, it would seem that the Society should study or perhaps undertake research on the following three fundamental questions which are given in order of importance.

1. The effects of prolonged exposure to low concentrations of CO_2 (0.06 to 0.20 per cent) from a consideration of possible respiratory and metabolic changes and changes in the pH of the blood.
2. The influence of relative humidity on the water balance of the tissues of the organism, nerve sensibility, irritability, etc.
3. Physical and physiological action of ozone in concentrations near and below the olfactory threshold.

A critical review of the literature on each of the subjects by members of the committee will indicate where information is inadequate and what additional research is needed.

[6] See No. 14 in List of Research Papers, p. 17.
[7] See No. 12 in List of Research Papers, p. 17.

5. DIRECT AND INDIRECT RADIATION WITH GRAVITY AIR CIRCULATION.—*Technical Advisory Committee:* H. F. Hutzel, *Chairman;* A. P. Kratz, H. R. Linn, J. F. McIntire, J. P. Magos, T. A. Novotney, R. N. Trane, G. L. Tuve.

Plans for the investigation of direct radiators and convectors have been completed and research has proceeded at the University of Illinois along the same lines as previously adopted, with tests in both the room heating testing plant and the warm-wall testing booth. The work has been limited due to the drastic reduction in the budget.

Tests with one type of non-ferrous convector, and the eupatheoscope, were made in the room heating testing plant to determine the effect of varying the relative humidity of the air upon the heat output. The relative humidity of the air during the tests made with the convector was varied from 6 to 68 per cent, and, although there was a slight tendency for the heat output to decrease as the relative humidity of the air was increased, the decrease with the relative humidity of the air at 68 per cent did not exceed one per cent of the heat output obtained with relative humidities of from 6 to 15 per cent. No appreciable change in the heat loss from the eupatheoscope occurred when the relative humidity of the air was varied over a range from 25 to 80 per cent.

The tests in the warm wall booth were continued for the purpose of consolidating all of the results previously obtained, and determining whether the heat outputs of all of the types of convectors tested could be represented by an equation of the form, $H = K\,(t_s - t_i)^n$; and whether some exponent, n, differing from 1.3 would be more representative of all of the types of convectors tested. The results of tests made with 24 convectors, consisting of seven different types, lead to the conclusion that the error resulting from the use of the exponent 1.5 in the correction factor is less than that resulting from the use of an exponent of 1.3 as now recommended in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code).

The results were presented at the Semi-Annual Meeting of the Society in a paper[8] published in the JOURNAL of the Society.

The work in the room heating testing plant now consists of:

(a) Effect of height of enclosure on heating effect;
(b) Effect of top outlets on heating effect; and
(c) Effect of recessing on heating effect and heat output.

The work in the warm wall booth consists of a study of the optimum inlet and outlet dimensions in relation to the free area through the heating unit, the gross cross-sectional area of the cabinet, and the cabinet height.

6. OIL BURNING DEVICES.—*Technical Advisory Committee:* H. F. Tapp, *Chairman;* Elliott Harrington, F. B. Howell, J. H. McIlvaine, L. E. Seeley, T. H. Smoot.

The investigation of the plans outlined for studies of oil burners at Yale University, in cooperation with the *American Oil Burner Association* and the Society, was dormant during the year because of lack of funds. A paper[9] resulting from earlier studies was presented at the Annual Meeting.

7. PIPE AND TUBING (SIZES) CARRYING LOW PRESSURE STEAM OR HOT WATER.—*Technical Advisory Committee:* S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, R. R. Seeber, W. K. Simpson.

The major work originally charged to this committee was completed in the laboratory studies made in Pittsburgh and reported some time ago. These studies gave

[8] See No. 5 in List of Research Papers, p. 16.
[9] See No. 2 in List of Research Papers, p. 16.

rather complete information on all the doubtful questions on flow of steam in pipe to which the committee gave consideration. The only additional work which the committee has in mind at present is the analysis of the various laboratory reports and the preparation of a final correlating report clearly setting forth recommendations for changes in pipe sizes resulting from the laboratory studies. A reduction in personnel of the Laboratory staff and the urge for other work have delayed completion of this study.

8. CORRELATING THERMAL RESEARCH.—*Technical Advisory Committee:* R. M. Conner, *Chairman;* D. S. Boyden, J. C. Fitts, H. T. Richardson, Perry West.

Plans for the correlation of research in industry, sponsored initially by S. R. Lewis, Chicago, Ill., as far back as 1927, have been continued to date. Although the original purposes of this activity have not been lost sight of, the work during the past few years has been on a rather restricted basis due to shortage of funds.

During the past year the cross-index card catalog file dealing particularly with thermal engineering subjects, has been maintained at the Society's Research Laboratory in Pittsburgh. Although this work has been financed on a greatly reduced basis, the various files have been maintained in satisfactory shape, and a great deal of useful information has been compiled for the benefit of anyone interested in thermal engineering subjects.

Lack of funds made it necessary for the committee to forego its plans for publishing a Directory this year. It is planned, however, to bring out the 1935 Directory of thermal research projects, and institutions participating in such work, prior to the expiration of the present school year. The committee feels that a very useful purpose can be served by placing at the disposal of interested trade associations and other scientific organizations this directory, showing the class of experimental work that is going on. It is believed that the proper use of such a publication should avoid useless duplication of effort and also assist in promoting more friendly relations and a better understanding among the various societies, associations, and other organizations comprising this branch of the industry. It is the committee's sincere belief that the objectives originally laid down and which have been followed so diligently during the past few years, should be continued to the end that useless duplication of effort in research, codification and standardization may be avoided and the best interests of the profession and of industry in general be served.

9. HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION.—*Technical Advisory Committee:* F. B. Rowley, *Chairman;* H. F. Bain, H. F. Hutzel, W. G. King, A. P. Kratz, E. J. Lindseth, L. P. Saunders, G. L. Tuve, W. E. Stark.

At the June meeting of the Society, Professor Tuve and Professor McKeeman presented a paper [10] covering the important results of the study carried on at Case School of Applied Science in cooperation with the Research Laboratory. Arrangements have since been made whereby the Mechanical Engineering Laboratory at Case School will provide certain equipment and time for continuing the work. Due to financial limitations, however, only a very modest rate of progress will be possible.

The above paper included the following:

1. A study of the methods for calculating the heat transfer of fin-tubing.
2. Results of experimental work on one type of fin-tubing, in which the effects of air velocity and of turbulence in the air stream were determined.
3. Correlation of these results with those from previous investigations, both American and European, with a table giving a summary of proposed methods for calculating the heating and cooling of air by forced convection.

Aside from air velocity and turbulence, the main factor affecting the heat transfer coefficient is the design of the fin-tube. Accordingly, there is now being constructed a new experimental unit for conveniently testing several kinds of fin tubes. The

[10] See No. 4 in List of Research Papers, p. 16.

arrangement will be similar to the unit previously studied with draw-through air flow, but the working section will be 16 in. square (instead of 35) and the heat input will be from a specially constructed electric boiler, with recirculation of the condensate. The input will therefore be measured electrically. One of the main features will be easy interchange of the test sections.

The committee is attempting to obtain a number of sets of samples of fin-tubing, so that the study may be extended to as many variations in design-proportions as possible. The work will be confined to a single bank of tubes for the present and therefore if twelve tubes of each design could be obtained, test sections could be constructed. The tubes will be mounted vertically, in parallel, with distributing orifices in the intake manifold. Tests will be made over a range of at least 500 to 1500 fpm linear air velocity through the free area. Possibly, manufacturers would be willing to loan completed test sections of this size (16 in. square area of fins).

With this set-up there will be investigated the variables of: (1) fin spacings per inch, (2) fin thickness, (3) fin height, (4) tube diameter, and (5) type of fin and fin attachment. If the test sections are obtained without too much delay, it should be possible to produce a reasonable amount of good data in time for presentation at the 1935 June meeting. These results should be applicable with fair accuracy to a wide variety of types of extended surface units, and should therefore be of considerable value to designers in the heating and air conditioning fields.

10. REFRIGERATION IN RELATION TO AIR TREATMENT.—*Technical Advisory Committee:* A. P. Kratz, *Chairman;* E. A. Brandt, John Everetts, Jr., E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.

The work in summer cooling as finally outlined by the committee involved the use of a two-ton mechanical refrigerating unit in connection with the forced air heating system employed for winter heating. For part of the season the artificial cooling in the daytime was supplemented by cooling with outdoor air at night. The outdoor air was circulated by means of the fan in the forced air system and the second story windows only were opened at night. For part of the season the windows were not opened and the artificial cooling was not supplemented by cooling with outdoor air at night. The machine had a constant output and the amount of cooling was controlled by *on* and *off* operation of the compressor to maintain a constant temperature of approximately 80 F in the house. Over the entire period requiring artificial cooling the fan was operated during both the *on* and *off* periods of the machine, and auxiliary outdoor air equivalent to approximately one air change per hour was taken into the house for the purpose of ventilation. Some tests were run with night cooling alone, in which case the air from outdoors was circulated by means of the fan in the forced air system and the second story windows only were opened.

The results of the summer's work indicated that in milder weather some saving was effected by supplementing the artificial cooling with night air cooling. In severe weather, however, the nights did not cool sufficiently to render such supplementary cooling effective. Night cooling with the fan in the forced air heating system circulating approximately nine air recirculations per hour and with the second story windows only opened was not nearly as effective as it was when the attic fan was used under the same conditions drawing in approximately 36 air changes per hour through the second story windows. The machine actually developed approximately 2½ tons of refrigeration and this was sufficient to maintain 80 F in the house even in the hottest weather. While the number of degree hours above 85 F for the season was about normal, the summer was unusually severe in that there was a high concentration of days having maximum temperatures exceeding 90 F. The results were presented in a paper [11] at the Annual Meeting of the Society. The *National Warm Air Heating and Air Conditioning Association,* and the *Utilities Research Commission* of Chicago cooperated in this study.

[11] See No. 13 in List of Research Papers, p. 17.

11. ATMOSPHERIC DUST AND AIR CLEANING DEVICES (Including Dust and Smoke).—*Technical Advisory Committee:* H. C. Murphy, *Chairman;* J. J. Bloomfield, M. I. Dorfan, Philip Drinker, Leonard Greenburg, S. R. Lewis, T. W. Pangborn, F. B. Rowley, Games Slayter.

The Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, which was formulated by the committee, was approved by ballot vote of the membership of the Society and adopted at the June meeting. The Code has also been submitted to the *American Standards Association* for acceptance as an Approved Standard with the sponsorship of the Society and approval of a number of engineering societies.

The Committee has in preparation a Code for Testing and Rating Air Cleaning Devices Used in Public Health Work. Considerable study has been devoted to existing procedures in handling hazardous dusts and the Committee hopes to have a tentative Code ready for discussion within the next few months. The formulation of such a Code is becoming increasingly important. Recent court decisions placing the responsibility for various industrial diseases squarely on the employer has resulted in the considerable elimination of dust hazards. The situation is made more acute because of the legal racket which is developing hundreds of suits at common law for injury or death due to silicosis and similar disorders. There are between 1000 and 1500 suits in New York State alone asking damages in the aggregate of $50,000,000.00.

At present there are no recognized standards for testing and rating dust arrestance of air cleaning devices used in dust hazardous work. The committee is working earnestly to develop such a Code in view of the commercial importance of the subject. The committee has also cooperated with a number of individuals and associations interested in the control of atmospheric pollution.

12. CORROSION.—*Technical Advisory Committee:* J. H. Walker, *Chairman;* H. F. Bain, E. L. Chappell, W. H. Driscoll, R. R. Seeber, C. M. Sterne.

The committee has had under consideration for the past two years a study of the effects of the chemical characteristics of condensate returning from a steam heating system on corrosion of iron and steel pipe. A program for the study outlined by the committee is being studied by Prof. R. R. Seeber of the Michigan College of Mining and Technology in cooperation with the Laboratory. The study is an unusually tedious one and progress will be made slowly. The committee feels, however, that definite progress toward a final solution is being made. The study to date has resulted in a progress report [12] presented at the last Annual Meeting of the Society.

The committee has also given consideration during the past year to a study of corrosion of copper and copper bearing alloys used in steam heating, ventilating and air conditioning systems. A general program for this study was developed, which it is hoped manufacturers will support.

13. GAS HEATING EQUIPMENT.—*Technical Advisory Committee;* W. E. Stark, *Chairman;* Robert Harper, E. A. Jones, Thomson King, J. F. McIntire, E. L. Tornquist, H. L. Whitelaw.

Work previously outlined by this committee is being continued.

14. AIR CONDITIONING IN TREATMENT OF DISEASES.—*Technical Advisory Committee:* E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, Margaret Ingels, H. R. Linn, E. L. Stammer.

This is a new committee appointed this year to give consideration to a program

[12] See No. 1 in List of Research Papers, p. 16.

for possible research dealing with the relation between atmospheric conditions and health, with particular reference to the effect of air conditioning in the treatment of diseases. Certain difficulties have arisen in instituting worthwhile investigations on the treatment of disease by air conditioning, due to the indefiniteness of certain terms involved, and to the magnitude of the task as a whole. What is air conditioning? What is treatment? And what diseases are affected by the air environment? To avoid an interminable discussion, let us say that air conditioning is the modification and control of our natural air environment in order to make it more suitable for the uses of mankind. This is a much broader definition than the simple control of temperature, humidity, air motion, and air cleanliness, which is favored from a commercial or engineering standpoint by some members of the Society.

But when we get into the subject of the treatment of disease, it becomes at once apparent that these four primary factors are entirely inadequate. For example, the work of Grollman of Johns Hopkins in the treatment of anemia by varying the air pressures comes clearly within the scope, or at least the possibilities, of air conditioning.

In our letter to the members of the Committee on April 19 we discussed the subject of classification. We stated in that letter that we had discarded the United States Census classification on morbidity and had tentatively adopted Dr. Wm. Osler's classification of disease under eleven different headings. We find, however, upon further study that this leads us into extensive investigations with questionable results, and it appears desirable therefore to throw out all of these accepted classifications of disease and adopt one of our own. We submit the following classification of diseases affected by the air environment:

Internal

1. Animal Parasites
   - —Malaria
   - —Yellow Fever
   - —Malta Fever, etc.
   - —Bites from Flies and Insects
2. Respiratory Diseases
   - —Colds
   - —Influenza
   - —Bronchitis
   - —Tuberculosis
   - —Pneumonoconiosis
3. Allergic Diseases
   - —Asthma
   - —Hay Fever

External

4. Thermal Diseases and Disturbance
   - —Sunstroke
   - —Heat Prostration
   - —Cold Exposure Reactions
   - —The Skin Temperature and Metabolism
5. Air Pressure Reactions
   - —Anemia
   - —Chlorosis

This classification, obviously, does not include a vast number of diseases which are affected by our air environment. For example, recent investigations indicate the possibility that constitutional diseases, heart disease, arteriosclerosis, nephritis, etc., may be very closely related to dust inhalation. The late Dr. Fenton B. Turk, in his investigation of shock and similar phenomena, developed his theory founded on a mass of experimental evidence that autolyzed body cells liberate a substance which he named cytost that is extremely toxic to homologous animals. A small piece of skin from the neck of a cat, for example, digested in sterile water and reinjected into the host will cause its death, in as short a period of time as three minutes. He pointed out that human beings in inclosed spaces are constantly giving off dead cells which in dusty air are breathed, possibly producing in this attenuation tissue changes and grave constitutional disturbances.

If we consider in our research program the obscure effects of recent research of this character the investigation would have no end, even though this work is of the greatest interest and probably of inestimable value to mankind.

In the first division of the suggested classification we have malaria, yellow fever, and other diseases caused by the bite of flies and insects, or by parasites introduced in this manner. We can dismiss these diseases from consideration with the understanding, of course, that they are easily eliminated by that state of air cleanliness which prevents their entrance into habitable rooms.

Under respiratory diseases the Committee will find much to investigate. The last item in this division, the pneumonoconiosis, includes silicosis, cholacosis, etc.; in fact, a large part of present activities in industrial hygiene.

Much work has already been done in the third classification, the Allergic Diseases. We mention only asthma and hay fever, but there are many others and the proper filtration of air has already opened up a valuable field for air conditioning.

Under External Effects we have subdivisions 4 and 5 relating to the reactions of the human body from variations of temperature, humidity, and air motion, and also from variations in air pressure. Research work in this particular field should produce immediate results, and we suggest first that with the new conception of skin temperature as determining human comfort, and with preliminary work already done on this subject by Emma France Ward of Johns Hopkins University, and our own research staff at Pittsburgh, we are now in a position to check effective temperature lines against the skin temperature of human subjects. This should be undertaken without delay.

Reviewing the preceding comments, and particularly the classification of diseases where air conditioning has a very definite therapeutic effect we suggest that work be first undertaken under the fourth classification. This deals with the effect of air temperature, humidity, and motion, on the human body.

One of the members of our committee, N. D. Adams of Rochester, Minnesota, is in close touch with many of the medical and surgical attendants at the Mayo Foundation. Through his intimate acquaintance with these men, many of whom enjoy an international reputation, we have been enabled to discuss some of the physiological and pathological problems involved in this study. Dr. Walter M. Boothby of that institute has made a painstaking study of oxygen therapeutics and has contributed considerable material of great value to the profession from his research work and clinical experience.

Dr. Sheard, Dr. Brown, Dr. Plummer, and others have interested themselves in our research problem and after several visits to the institution by the writer, and the Chairman of the Research Committee, Mr. Howatt, a program was suggested whereby the A. S. H. V. E. Research work could be carried on in this institution, particularly along the lines to be developed in the fourth classification of diseases. The institution is exceptionally well equipped with apparatus of every kind, oxygen rooms, skin temperature devices, etc., and a study here of both the normal and abnormal would be of inestimable value not only to our Society from an engineering standpoint but to the medical profession from a therapeutic standpoint.

15. THE EFFECT OF ENTERING TEMPERATURE AND VELOCITY ON THE TEMPERATURE AND DISTRIBUTION OF AIR WITHIN AN ENCLOSURE.—*Technical Advisory Committee:* C. H. Randolph, *Chairman;* E. H. Baars, John Jung, F. A. Kartak, J. E. Schoen, Ernest Szekely, J. H. Volk.

The objective of this committee is to determine the effect on air distribution and temperature variation within a ventilated space of the entering temperature, velocity and direction of the air, and the location and type of the grilles through which the air is admitted.

Plans for the work have been partially completed and a cooperative program has been arranged to be carried out at Marquette University in Milwaukee in cooperation with the Research Laboratory. Equipment is being installed whereby conditioned air can be supplied to a room. The air conditioning and control equipment will include water cooled surfaces supplied with water from a large tank and Baudelot coil, and refrigeration for cooling the water, supplied by a 3-ton ammonia compressor. Additional control and measuring equipment is being installed in order to give complete operating control of test conditions.

16. MINIMUM TEMPERATURE AND METHOD OF INTRODUCTION OF COOLING AIR IN CLASSROOMS.—*Technical Advisory*

Committee: Perry West, *Chairman;* J. D. Cassell, S. R. Lewis, J. R. McColl, A. J. Nesbitt, G. E. Otis, C.-E. A. Winslow.

This study aims to establish the relation of the temperature and other characteristics of the air discharge into a room by a unit ventilator to the prevalence of drafts. During a large part of the time, even during the heating season, proper school ventilation is a cooling rather than a heating problem. It has been well demonstrated that body heat given off by the children is sufficient to maintain proper room temperature with some minimum outside temperature which may range as low as 40 to 50 F, with no sunshine. For rooms on the sunny side of the building the combined heat source from children and sunshine is sufficient to maintain the desired temperature indoors with considerably lower outside temperature.

When cooling is required, it may be had with a unit ventilator either by bringing in the proper volume of outside air at the prevailing outside temperature, a larger volume of air with the addition of some heat in order to give a desired temperature to the air blown into the room, or by the proper volume of outside air at the prevailing temperature mixed with sufficient re-circulated air, in order to give a desired temperature to the air blown into the room. These possibilities make desirable a study to determine the minimum satisfactory temperature at which air can be so admitted into a classroom without giving the effect of cold drafts on the children in some portions of the room, and also the quantity of outside air in cfm per pupil that must be brought in under each of the three above mentioned conditions for various outside air and weather conditions and various classroom exposures.

It is admitted that a wide variation of low temperatures of air admission may theoretically be possible with an air jet into the room having proper velocity, location and direction for any particular room in question. However, it is obvious that with units designed and built in a factory for installation in any classroom under a wide variety of conditions there must be some minimum temperature of air admittance below which freedom of cold drafts can be insured under all practical conditions of installation, operation and room design. It is this minimum which it is proposed to have established by the investigation.

Prof. G. L. Larson, Madison, Wis., moved that the report be filed and that Mr. Howatt be given a vote of thanks for the fine work he had done as chairman of the Committee on Research. The motion was seconded and carried.

On Tuesday morning President Haynes called the meeting to order and a report of the Committee on Garage Ventilation was presented by E. K. Campbell, Chairman. He gave a brief review of the work that had been done by the Society since 1925 and of the cooperative efforts undertaken with the *National Fire Protection Association* which resulted in the development and adoption of a Code issued by the National Board of Fire Underwriters, that is now used by insurance companies.

COMMITTEE ON VENTILATION OF GARAGES AND BUS TERMINALS

REPORT OF JANUARY, 1935

The matter of garage ventilation has been before the Society so many times during the course of this Committee's work that it is difficult to know just how to discuss the subject and how much history to include in order that everyone present shall understand our recommendations. However, back in 1925 the Society appointed a representative to serve on a committee of the *National Fire Protection Association* to assist in the formulation of fire insurance regulations for the construction, heating and ventilating of garages. That work was successfully concluded by the development of a Code which was finally adopted by the insurance world and published in

Pamphlet No. 88 of the standard publication of the *National Board of Fire Underwriters.*

The Code as adopted by this Society will be found in A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929. The main premise of the Society's Committee has been generally accepted, namely that the danger of fire in a garage lies in the formation of pockets of explosive gases and not that all of the air of the garage might become explosive.

The Fire Prevention section of the Code here submitted is based on that premise and the text follows very closely the wording of the insurance regulations.

The first sections relate entirely to fire prevention, and the carbon monoxide section has been added by your Committee.

When our work with the *National Fire Protection Association* was completed, the Council instructed the Committee to investigate the matter of garage ventilation from the standpoint of carbon monoxide. Comparatively little data were found so we began seeking cooperative agreements to get the necessary data on which to base a suggested method of ventilating.

The first series of investigations were made in a bus garage in connection with Washington University at St. Louis, and the only results of these studies were to the effect that a bad condition existed. We then tried to carry the investigation to Boston but were unable to get anything done there. We finally succeeded in getting the Society's Research Laboratory to undertake some investigations in Pittsburgh.

First an underground ramp storage garage was investigated and it was found that there was almost no problem existing in the ordinary storage garage. There was no problem in that particular garage because they had more ventilating apparatus than they needed. Lacking definite information to follow, they had provided ample capacity in view of the tremendous problem involved in a garage which is five stories underground.

A one-story repair garage was also investigated and there it was found that the best method of introducing air was one which diluted directly the lower strata of air in which the men worked, but that test was made when heating was not required, and the question immediately arose, "What would happen if you had to introduce heated air?" Of course you must introduce heated air when the weather is cold or there will be no ventilation. Ventilation is not going to be provided at the expense of human comfort. Men simply will not stand for it. Another test was made in the same garage as to the best method of introducing the air, and it was found that the method recommended accomplished the desired results with about 20 per cent less air heated and forced into the building than by any other method.

The committee also carried on work in connection with the University of Kansas, some in Lawrence, where the University is located, and some in Kansas City, and it was found that the monoxide problem was encountered in connection with either very large garages or in various types of service garages.

All of the strictly storage garages which were tested presented little or no problem, but the service garages, where engines were tuned or carburetors were adjusted, were the ones where the problem became acute.

In one garage we investigated in Kansas City, and put in an experimental installation to correct their difficulty, there was one small section on a raised floor where some men worked repairing fractional electric motors. They were not connected with the garage but they were in the same room, and the carbon monoxide from the cars drifted up into that section.

Because of the carbon monoxide present in the air, the men were taking aspirin and sometimes had to be taken home before the day's work was finished.

The experimental installation based on the theories which we have set forth handled the situation perfectly, so that the garage owners would not permit the installation to be taken out after it had been demonstrated.

The carbon monoxide section of the Code provides a permissible standard of not more than one part carbon monoxide in 10,000 parts of air, where men are working 4 or more hours per day.

The question has been raised as to the reason for that severe restriction. The Committee did not attempt to go into the physiological aspect of this question at all, but took other work which was well authenticated as the basis for that standard.

Compared with the standard in the Holland Tunnel, where it is four parts in 10,000, it was of course quite severe, but the Holland Tunnel is based on a person being in there not more than 30 min, and even the traffic policemen are not allowed to stay on duty more than 2 hours. Hence this is an entirely different problem from one in a garage, where men are at work for 8 hours.

Then, as Director Houghten has pointed out in his paper, one of the great problems was how to devise a code that would be generally acceptable and could be made effective without great expense for installation and operation.

The great need was for some ready means of indication of what the condition in the garage might be. The installation of a very complete ventilating system could be required, but with no method of detection of carbon monoxide except the headache method, the operation might be so expensive that garage owners and operators would refuse to use it at all. Hence, we searched extensively for some sort of an instrument within reach of the ordinary garage owner in order to guide him in the operation of his ventilating system.

The Mine Safety Appliances Co. finally became convinced of the need for such an instrument and has developed the one recently tested, so that we recommend in paragraph B that this instrument be used as a guide. It is not a perfect instrument and it won't measure exactly the one part in 10,000, which is specified, but it is the best, and as far as I know, the only thing that is available at the present time.

In paragraph C we specify the standard for the installation of the system. The variation in the amount of carbon monoxide emitted by the different cars is very great, hence such a figure can only be an average. As the ventilating system can be operated to maintain an air quality that is constantly checked, there is no material disadvantage of installing a plant of ample capacity.

It will now be possible for a garage owner to have a ventilating system that will definitely take care of a liberal number of idling cars. Each building is a separate problem and its requirements must be determined. It must be borne in mind too that the quantities specified in the Code are not in addition to those required for fire prevention, but the quantities required for fire prevention are part of that required for dilution of carbon monoxide or vice versa.

The recommended method of introducing the air is to have it directed to the floor. In the Pittsburgh test to determine the best method of introducing heated air, it was found that by bringing the air in through ducts overhead, and driving it downward vertically into the aisles where cars do not stand for any considerable length of time, the air could reach the floor, then would spread in a cone across the floor, mix with the lower layer of air and dilute that in the area of the garage which was in use. The same thing could be done with units having outside air connections.

All of the tests indicated the economy and effectiveness of the method when used in connection with a heating system.

The final requirement of the Code, namely that the air for normal dilution must be heated to 50 F when it is brought in for ventilating purposes, means that ventilation is not to be provided at the expense of human comfort. If ventilation could be provided without regard to comfort, open windows would meet almost any condition except in an underground garage. With any heat in the garage, an open window in cold weather provides a tremendous air circulation and experience has shown that a fan which brings in cold air will not be used except for short periods and in case of great need.

In order to get effective ventilation, the air must be heated, and heating capacity should be provided to heat the normal volume of air necessary to dilute the carbon monoxide in the garage. The air must be heated to at least 50 F.

As it is not always possible to calculate in advance the number of cars which may be idling in a garage and because it is difficult to persuade owners to provide heating capacity to take care of the necessary air volume, the Code does say that "additional air from outside not necessarily heated to 50 F should be provided for emergency conditions." Your Committee does not regard this particular clause as desirable but thinks that this concession to the economics involved is necessary under some conditions, to have sufficient ventilation provided even though the air is not heated.

The Code in its present form represents work done during a period of 10 years by the Committee and it is believed that it is in form for adoption by the Society and the request is made that this Code be presented to the Society for vote by the usual letter ballot.

Respectfully submitted,

E. K. CAMPBELL, *Chairman*,
S. H. DOWNS, F. H. HECHT,
T. M. DUGAN, H. L. MOORE,
E. C. EVANS, A. H. SLUSS,

COMMITTEE ON VENTILATION OF GARAGES AND BUS TERMINALS.

Code of Minimum Requirements for Heating and Ventilating Garages

(*Revised Issue of January, 1935*)

12. BASEMENT AND SUB-BASEMENT GARAGES

a. Basement and sub-basement garages shall be continuously ventilated by a mechanical ventilating system with positive means for both the inlet and exhaust of at least 1 cu ft of air per minute per square foot of floor area. Control of either the exhaust or inlet fan shall be close to the entrance door. The ventilating equipment may be combined with the heating system which is governed by Section 22.

b. Each basement and sub-basement garage shall be completely protected by an approved system of automatic sprinklers.

c. No repairing of any kind shall be made in any basement or sub-basement garage.

d. Each basement and sub-basement garage shall be provided with exits in the ratio of one for every 2500 sq ft of floor area, provided that the number of exits shall never be less than two.

e. There shall be no facilities for gasoline handling or filling operations in any basement or sub-basement garage and no such filling or handling operations shall be permitted therein.

22. HEATING

a. Heat generating apparatus, other than direct-fired unit heaters conforming to paragraph c, shall be placed in a separate room used for no other purpose and cut off from all other parts of the building by construction equivalent to 8-in. brick walls and 4-in. reinforced concrete floors and ceilings. Openings in the above mentioned cut-offs shall be restricted to those necessary for heating pipes and ducts. Entrance to room containing the heating plant shall be from the outside only. Heat generating plants shall preferably be located in a detached building.

b. Heating systems employing recirculation of air, other than unit heaters, shall also comply with the following:

1. Positive recirculation of at least 1 cu ft of air per minute per square foot of floor area within the garage shall be provided in all garages having an average ceiling height of 15 ft or less; in garages having greater average ceiling heights the volume of air recirculated shall be increased in proportion to the increase in height.

2. Recirculating systems drawing air from the floor to the fans and discharging the air horizontally above the cars, shall have the intake openings not more than 2 ft above the floor measured to the top of the openings. Recirculating systems which drive the air down to the floor vertically, may take the return air to the fan from the fan level.

3. Not less than 5 per cent of the air moved by the fan shall be taken direct from outside of the building through a duct which shall deliver the outside air direct to the fan or to a point near a fan which rests on the floor; the duct shall be open at all times and the air supply which it provides shall be without control. This provision does not set aside the ventilating requirement of section 12a for basement and sub-basement garages.

c. Direct-fired unit heaters shall be of a type listed by Underwriter's Laboratories for use in garages, airplane hangars and/or similar locations and installed near the

eave level or at least 8 ft above the floor. A safe distance shall be maintained between the heater and adjacent combustible material in conformance with the Regulations for the Installation, Maintenance and Use of Pipes and Fittings for City Gas, which specify that continued operation of heaters shall not raise the temperature of surrounding combustible material above a maximum of 160 F.

d. Gravity warm air heating systems shall not be used.

e. Fans used in either heating or ventilating for recirculating or exhausting air from the garage, which are not at least 4 ft above the floor, and fans drawing air from within 4 ft of the floor shall be of the non-sparking type.

23. VENTILATION

a. The following garages shall be provided with means for the ventilation as needed and made necessary from time to time by operating conditions, and in compliance with the provisions hereinafter set forth:

1. Garages having a capacity of 6 to 24 motor vehicles and having less than two walls with window openings;
2. Garages having a capacity of 25 to 34 motor vehicles and having less than three walls with window openings; and
3. All garages having a capacity of more than 34 motor vehicles.

b. Natural ventilation may be employed where practicable to maintain open windows or other openings at any time. Such openings shall be distributed as uniformly as possible in at least two outside walls and shall have a total area at least 5 per cent of the floor area. Where it is impracticable to operate such a system of natural ventilation, a mechanical ventilating system shall be provided.

c. All mechanical ventilation systems, which may or may not be combined with the heating system, shall comply with the following provisions as a minimum requirement for fire prevention:

1. Positive provision shall be made for the inlet of at least one cubic foot of air per minute from out of doors for each square foot of floor area, or for removing the same amount and discharging it to the outside.
2. Positive means of handling air refers to power driven fans and also includes, for garages of not over 35 vehicles capacity, air exhaust stacks, provided they have not less than 15 sq ft of steam heating surface for each square foot of duct area, and not less than 1 sq ft of free area through both heating coil and duct for each 350 sq ft of floor area. Such an exhaust duct shall discharge above the roof and extend in any case to a height of not less than 15 ft above the heating coils.
3. Where positive systems of exhausting air are used, the exhaust openings shall extend not more than 2 ft above the floor and shall not be more than 50 ft apart, around the perimeter of the garage.
4. Where positive systems of introducing outside air are used and where air is recirculated the air may be delivered horizontally or it may be driven vertically to the floor between cars but in either case it shall be in sufficient volume, in sufficient number of locations and with sufficient velocity to secure distribution to all parts of the building. The height of the air inlet opening for horizontal discharges shall be such that the air will be discharged above the top of the vehicles. The location of the downward vertical discharges shall be such that the air will be discharged downward between vehicles and with sufficient velocity to reach the floor.

d. All duct openings, either supply or exhaust, shall be covered with ¼ in. mesh screen which shall be kept free from obstructions.

e. The passing of air ducts through fire walls shall be avoided. Ducts shall be installed in accordance with the Regulations for the installation of Blower and Exhaust Systems, as published by the *National Board of Fire Underwriters.*

24. REPAIR WORK

a. Garages intended primarily for the storage of motor vehicles should be kept as free from the hazards of repair work as possible. This is particularly important where the vehicles stored are of high value as in bus garages. It is desirable that such processes as carbon and lead burning, and welding and cutting, be separated from storage sections as indicated in the following paragraph. If, as in service stations, some vehicles must be stored in the repair sections, the number of such vehicles should be kept as small as practicable and if possible separation by partitions of at least 1-hour fire resistance provided.

b. Repair work involving open flames, spark emitting devices or highly heated

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parts should be done in a section that is separated from the storage section by a standard fire wall if the building is of other than "fireproof construction" and is not protected by an approved system of Automatic Sprinklers. If the building is of "fireproof construction" or is protected by an approved system of automatic sprinklers, the repair shop may be cut off from the storage section by fire partitions.

c. Such repair sections shall be equipped with a mechanical ventilating system with provision for both the inlet and exhaust of at least 1 cu ft of air per minute per square foot of floor area. It is recommended that in engine testing, the exhaust be discharged direct to outdoors through a duct or pipe of incombustible material and suitable size, installed as an extension of the exhaust pipe or muffler, in which case either the mechanical system for inlet of air or mechanical system for exhaust may be omitted.

d. Where standard automatic sprinkler protection is not provided, paint, carpenter and upholstery shops shall be cut off from the storage section by standard fire walls in non-fireproof buildings. Where such sprinkler protection is provided or the building is of "fireproof construction" the cut-off walls may be fire partitions. In all garages, paint, carpenter and upholstery shops shall be cut off from the repair shop by fire partitions.

e. The arrangement, construction, and ventilation and protection of spray booths for paint, varnish or lacquer spraying and the storage and handling of these finishing materials shall be in accordance with the Regulations on Paint Spraying and Spray Booths.

f. In carpenter shops, the hazard of dust explosions shall be guarded against by the installation of dust collecting systems when deemed necessary by the authority having jurisdiction. (See Regulations on Blowers and Exhaust Systems.)

g. Steam generators for tire vulcanizing, water heaters, and other fuel burning appliances such as forges shall not be installed within the storage section of a garage.

25. INSPECTION AND REPAIR PITS AND TRESTLES

a. Elevated trestles or hoists are preferable for this service. If pits are used, they shall be continuously ventilated by a system independent of the main garage ventilating system. Such pits shall be cleaned at least daily and no accumulation of oil and grease permitted. Permanent illumination shall be provided, but this shall not be construed as prohibiting the supplementary use of approved, properly guarded portable lamps attached by approved, heavy, reinforced extension cord.

b. When pits are constructed with one end open to an outside wall, either directly or through a transverse pit, at the same or lower level, natural ventilation may be permitted provided floor level of pit is above natural outside grade, with air inlets at pit floor level opposite each pit. There shall also be a ventilating duct at closed end of each pit; such ducts shall extend to the outside air and be equipped with an approved suction creating device.

CARBON MONOXIDE HAZARD

This section shall apply to the same class of garages as Section 23, all of the provisions of which remain in force and are supplemented by the following provisions:

a. The standard of carbon monoxide dilution within a garage in which men are employed four or more hours a day, shall be not more than one part of carbon monoxide in 10,000 parts of air.

b. Wherever the carbon monoxide hazard is known to exist, it is recommended that an indicating instrument or alarm be used which may be connected up as an automatic control, in which case the ventilating system can be operated in part or in whole as is shown necessary by the indicating instrument.

c. It is recommended that including the air specified under Sec. 23 there shall be introduced into the garage a minimum of 5000 cfm from out of doors for every passenger car with engine idling in the garage, and 10,000 cfm for every truck or bus.

d. The recommended method for introducing the air is with downward discharge with sufficient velocity so that the air strikes the floor and spreads, mixing with and diluting the lower strata of air in which the men are working. This arrangement

may be accomplished either by ducts from a central fan and heater, or by units having connections so that air may be taken from out of doors.

e. Where the method recommended in paragraph d is used, vents for removing the carbon monoxide laden air in equal volume with the air brought in from outside, shall preferably be located at or near the ceiling.[13]

f. The heating system must provide sufficient heat to raise the temperature of all air admitted for ventilating purposes under normal conditions to not less than 50 F and the heating system must maintain the specified temperature in the garage when the normal ventilating system is in full operation under extreme weather conditions. Additional air from outside not necessarily heated to 50 F should be provided for emergency conditions.

DISCUSSION

MR. THOMPSON: I would like to ask whether the ordinary garage man will spend the necessary money to provide the extra heat necessary for the required additional ventilation.

Also, is there any objection to the use of ozone in a garage to reduce the amount of ventilation required by the fire restrictions, and can ozone be used to directly reduce the amount of heating and ventilating required for occupancy?

H. L. ALT: I wish this report had been available about two years ago. At that time I had a problem of a large post-office driveway in which it was intended to have 100 motor trucks as a maximum occupancy. The supposition was that the drivers shut down their motors when they came in and backed up to unload or load mail, but nobody was able to guarantee it. The driveway came in from a side street, turned at a right angle, and went out to an avenue, with loading platforms on each side.

I made a long study of the Hudson Tunnel experiments and found a great variety of CO emissions for acceleration and deceleration at various miles per hour, but did not locate any data on idling trucks. Finally, with a lack of any other information, it was assumed that the air circulation requirements should be one cubic foot per minute, simply because an ordinary pleasure car gave off somewhat less than that amount in the previously recorded tests. Then two parts of CO in ten thousand were taken as the maximum that could possibly be maintained with men working in this space 8 hours a day.

We were forced by the physical conditions to take part of this air off at the ceiling and part at the floor, roughly 50 per cent in each case. The air supply for the driveway comes in at two doors, and the ceiling has perhaps 75 unit heaters which are directed into this air to temper it.

I would like to ask how that would compare with the experiments made so far in garage ventilation. In view of our present information, is that a fairly good percentage of CO_2 for that condition, or should it be increased or diminished?

E. K. CAMPBELL: In answer to the question whether the ordinary garage owner can afford to install and operate a system of this design, it is of course true that if anyone else can develop a system that will do the same work as effectively at less expense it should be considered. However, we know of no way to reduce the cost of a system which will effectively handle the CO problem, and if we are to recommend a design Code it must be a system that will actually work.

Ozone was one of the first things that we investigated. We found that it has little or no effect on carbon monoxide. Its introduction into a garage might possibly

[13] Carbon monoxide as part of the exhaust gases of an automobile are heated to a high temperature and rise, and with outside air heated and delivered to the floor, tests have shown that approximately 20 per cent less air is required for a given dilution of the carbon monoxide in the occupied zone than on other tested arrangements. See paper, Carbon Monoxide Distribution in Relation to the Heating and Ventilating of a One Story Garage, by F. C. Houghten and Paul McDermott (A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933).

have a little physiological effect on the men at work there, enabling them to throw off the carbon monoxide more effectively, but that question is open to debate. At the present time, ozone does not enter into the problem of diluting or reducing the carbon monoxide content in a garage.

Mr. Alt's question as to how two parts of CO in ten thousand compare is answered by the Code itself. That is, we recommend one part as a desirable standard. My personal opinion is that men working in air which contains two parts of carbon monoxide for any length of time will develop headaches.

W. L. FLEISHER: Some years ago I was working on garage ventilation, applying practically the same method of determining CO_2 which has been developed by Mr. Houghten in the laboratory. Realizing that at certain times a much larger amount of outside air would be required than at others, and that to heat this air required the installation of a boiler plant which was out of all proportion to the normal requirements of garage ventilation, a system was developed which circulated the air in the garage in proportion to the normal amount of carbon monoxide that was given off. Through an instrument similar to that used by Mr. Houghten a damper was connected to the out-doors, which opened or closed to produce a larger quantity of out-door air than normally would be required. This damper was actuated by the control. During the critical periods of high monoxide concentration the proper amount of out-door air was obtained, not heated; in other words, during those critical periods the building was probably cooler than was ordinarily comfortable, but in sufficient quantities to be a protective measure.

In this way the boiler plant was only designed for the normal amount of air, and during those critical periods when danger might have arisen, the outside damper opened wide under the action of the monoxide indicator, giving the amount of undiluted outside air that was required for ventilation.

W. L. Fleisher, New York, presented the report of the Guide Publication Committee.

Report of Guide Publication Committee

The Guide Publication Committee, through its Chairman, brings a report covering the activities of the past year, that is full of optimism, justified we believe because of the future outlook and the progress made through the enormous amount of work that the Committee has had to devote to THE GUIDE during this period.

Before going into details, just for a moment please recall that due to the great financial embarrassment, not only of the Society but the country at large, that the 1934 Guide Publication Committee was handicapped by the absence of a technical secretary. When, in June of 1933, a technical secretary was dispensed with, the Guide Publication Committee, under my chairmanship, found itself with practically the whole GUIDE in front of it and nothing in shape for publication and no coordination of material. It was essential, therefore, for your Chairman to drop practically all his other activities and to devote most of his time from June, 1933, until the publication date of THE GUIDE 1934. The Guide Publication Committee of that year, and I think the Society, felt that these efforts were justified in that THE GUIDE 1934 was eminently successful not only from the standpoint of its technical data, but also from its general sales, the issue having been sold out in the early part of the fall of 1934. The commendations that resulted from this issue were generally favorable.

Your Chairman, however, during this period learned a great deal more about THE GUIDE and its possibilities than he had known before, although he had served as a member of the Guide Publication Committee for several years before that. When he was approached to assume the responsibility for THE GUIDE 1935, he felt that although for business reasons he should not assume this responsibility, nevertheless, he felt that THE GUIDE had such possibilities of service to the industry and such possibilities of adding materially to the support of the Society, that he reluctantly accepted the chairmanship for another year on the condition that the Committee work

should be divided among people absolutely competent to assume the responsibilities of the work assigned to them without very much assistance from the Chairman, who would outline and assign the work, but who was not supposed to do as much of the actual work as he had done for THE GUIDE 1934. Your Chairman also imposed a condition on his assumption of the chairmanship, which was that THE GUIDE should be designed and revised in such a way that no future chairman or no future committee, unassisted by paid technical assistance, should have the major duties which for the last two years, due to the financial condition of the Society, have been imposed on the Chairman, and that THE GUIDE should be put in a position to have an earning capacity that its set-up and its general trend had not presented before. In other words, the Chairman of your Committee felt that if THE GUIDE could, without in any way destroying its impartial attitude, make a greater appeal to the manufacturers, either through its circulation or because of the interest that could be generated in the younger generations, who would be future buyers, or by presenting something to the manufacturer that he could further broadcast, that THE GUIDE could be made into a publication which would not only support itself, but which would practically support the Research Laboratory of the Society.

In revising the chapters which applied to manufactured articles the Committee obtained the cooperation of engineers from the organized groups of manufacturers so that the text matter of THE GUIDE would represent the latest practice and developments. The thing of major importance that your Chairman was interested in was developing through capable assistants, unpaid it is true, but interested in the Society, problems in practice referring to the text of the chapter which would make an appeal to the schools and colleges and industrial concerns selling apparatus, so that the text embodied in THE GUIDE would have a new interpretation and would create comment and criticism which would lead to the normal and logical development of the text in future issues, based on the problems in practice which were presented to the Guide Publication Committee by outside sources. Your Chairman felt that these problems in practice, which are the major change in THE GUIDE 1935, could readily be changed from year to year, adding new problems, and adding new answers, so that without changing the fundamental text matter, which he was particularly anxious should remain undisturbed for a number of years until the various readers of THE GUIDE and those referring to it became accustomed to it, THE GUIDE, itself, would still be a new book each year simply by bringing the general data up to date and changing the problems and the answers.

When sufficient comment and criticism had been offered, or when sufficient time had elapsed for the interpretation of the chapter to have been thoroughly understood through the problems that were presented, then the text of the chapters, themselves, could be changed without the necessity of calling in outside parties to re-write them.

With all of these major problems in mind, your Chairman accepted his job and called a meeting of the Guide Publication Committee in Chicago in April, which meeting every member of the Committee, without expense to the Society, attended. Besides the actual Committee, your Secretary, who has been absolutely invaluable in this entire work, and your President also attended this meeting. After a full day's meeting, the plans that had been formulated were accepted and starting as early as April, it was assumed that the work of getting out the new GUIDE would progress very rapidly and be in good shape for early publication. However, due to the very major changes that this whole idea involved, and the fact that the aspects of THE GUIDE had to be so carefully considered, and that all of the Committee men were very busy men (outside of THE GUIDE work that they had so very generously assumed), by the end of June, when another meeting was held at Buck Hill Falls, work had not progressed nearly as rapidly as your Chairman and the balance of the Committee had hoped. It was essential, in order to accomplish the major changes that had been determined by us, to have the services of J. L. Blackshaw, who had been assisting Director Houghten in the Research Laboratory. Your Committee, working with Mr. Blackshaw, was able to get sufficient assistance so that at this meeting we can announce that THE GUIDE in its new form, with about 150 additional pages, is nearly ready for distribution and that in the mind of your Chairman, the "Problems in Practice" which are appended to every chapter so greatly broaden the information and scope of THE GUIDE, that all members of the Society will be able to look upon it with great pride, and use it with great value.

The enthusiasm of the Committee and the work of Mr. McDonnell and of your Secretary, Mr. Hutchinson, spread also to the advertisers and nearly 40 per cent increase in advertising is embodied in THE GUIDE 1935. We all feel that when this Guide, with its new set-up and increased advertising, is in the hands of the public and in the hands of the colleges and schools, that the following issues will not only be of greater service, but that the advertising will double or triple in the near future.

I give herewith a tentative estimate of receipts and expenditures of THE GUIDE 1935:

Income		
Guide advertising (1935)	$21,022.00	
Guide copy sales (1934)	11,349.09	
		$32,371.09
Expense		
Printing and Binding	$11,600.75	
Mailing	2,000.00	
Paper	2,447.52	
Engraving and Art Work	342.01	
Advertising Sales Promotion (1935)	1,651.73	
Copy Sales Promotion (1934)	2,480.69	
Editorial and Clerical	1,208.08	
		$21,730.78

Of course, all these things have been helped by the increase in the interest in air conditioning and although the chapters on air conditioning have not been too greatly changed except in the complete re-writing of such chapters as Unit Air Conditioners, nevertheless, the questions and answers appended to the chapters on air conditioning so broaden the scope of those chapters, that no one interested in air conditioning can afford to be without a copy of THE GUIDE.

Your Guide Committee will accept criticism or comment on all "Problems in Practice." In that way we hope to improve and perpetuate THE GUIDE for a much larger and more beneficial use.

In order to further the general interest in THE GUIDE and in the Society and to increase the revenues of the Society, your Chairman has proposed to the Council that some award be created for students in the schools and colleges to submit new "Problems in Practice," based entirely on the interpretation by an individual of the text of the chapter. The best questions and answers or the best group of questions and answers would then be published in the following issue of THE GUIDE and the person submitting the best set should be given as an award a fully paid trip to the summer meeting of the Society. Further, your Chairman has suggested that minor awards be made for other sets of questions and answers submitted.

A good many years ago, when your Chairman was doing certain work for Smith Brothers, the advertising agent of that company informed him that they distributed thirty million blotters a year to the school children of the country, so that when the children grew up, cough drops and Smith Brothers were synonymous in their minds.

Your Chairman feels that the making of such awards as have been outlined for the best questions and answers submitted, based on the text in THE GUIDE, will have a similar effect on the minds of the students in the schools and colleges of the country and that what they see or read in THE GUIDE during this formative period will make such an impression, that when they are full-fledged engineers or connected with companies purchasing any material advertised in THE GUIDE, they will think of those names and those things which have been impressed upon them during their formative period. In this way THE GUIDE becomes of greater importance to the advertiser and should increase the potential possibilities of THE GUIDE as an advertising medium enormously.

During the past year 8000 copies of THE GUIDE 1934 were distributed and sold. Feeling that THE GUIDE 1935 will be of much greater importance, we have authorized the printing of 10,000 copies for this year and think that there should be no difficulty in disposing of them. If we do, THE GUIDE will be a really very successful

financial proposition and will have justified all of the enthusiasm and all of the optimism which your Chairman and the Guide Publication Committee have felt.

There is only one other thing which has been advocated or developed by your Committee, which is the possibility of distribution of separate chapters by manufacturers, the manufacturer adding to the reprint of the chapter that advertising which he has published in THE GUIDE, itself. For instance, some of the unit air conditioning people having 1800 agencies and probably five times that amount of distribution, having prospective customers amounting to, maybe, a million, might, with profit, purchase from the Society large quantities of reprint chapters with problems and answers and their advertising attached thereto for free distribution to their prospective customers.

In this way we can increase the general use of THE GUIDE and the revenues of the Society. With all of these advanced ideas and possibilities of circulation and sale, there may be a question as to the reason for our not having printed more than 10,000 copies. The answer is that if more than 10,000 copies can be properly distributed, the advertising rates under which we are now operating are too low. If it seems possible from the acceptance of THE GUIDE 1935, that twelve or fifteen thousand copies of THE GUIDE 1936 can be distributed, then the Society must raise the rates for its advertising to substantially increase the income of the Society.

Your Chairman is anxious to bring up the point that he discussed with the Council last year, and that is the question of closer cooperation and a closer connection between THE GUIDE and the Research Laboratory. The Guide Publication Committee has used advantageously the research part of the Research Laboratory and in its chapters, THE GUIDE has incorporated those findings of the various research departments of the Society—that means the Pittsburgh work and all of the university findings that the Committee felt were well enough established to be incorporated in THE GUIDE. Of course, it must be understood that the Research Laboratory, being so badly handicapped for funds, has been unable to undertake quite a few of the important problems that have arisen during the last few years. This year your Chairman served also on one of the special technical committees, the one on Air Conditioning in Relation to Comfort and Health, and from this association was able to gather enough information to change and correct some of the text matter of THE GUIDE. It is for this reason that he feels that the Guide Publication Committee should be very closely allied to and associated with the technical advisory bodies of the Society, so that the findings of these various bodies and the Research Committee should find expression in THE GUIDE text or that THE GUIDE text should be edited to exclude those things that these various bodies find are not definitely enough established to warrant publication.

The Chairman wishes to bring out the fact that in the chapter on Coefficients of Transmission the cuts which have not been printed since THE GUIDE 1932 have been improved and altered and re-introduced into THE GUIDE 1935, and that the psychrometric chart has been re-drawn and republished as a separate sheet. As everybody working on heating and air conditioning, in which the total heat of the air is used as a function of design, has had so much occasion to use this chart, the Committee felt that even though it added to the expense of publication, it should be republished and revised in easily usable form. Many additional copies of this chart have been printed, and it is the feeling of the Committee and of your Secretary that to obviate the necessity of removing this chart from THE GUIDE, itself, that all the purchasers of THE GUIDE, or all those receiving THE GUIDE as part of their dues, should be given the privilege of buying detached copies at twenty-five cents as against the regular price of fifty cents which would be charged to the general public.

It is the Committee's opinion, and it is made as a suggestion, that the Society, through selected committees, should approach the various organized manufacturing groups with the idea of having them select and pay for someone who is sufficiently competent and skilled to write a new chapter covering the technical and practical side of the apparatus or material that they sell, and that such chapters, after being written by designated authors, should, of course, be reviewed by the Guide Publication Committee. As the manufactured articles employed in heating, ventilating and air conditioning are changing rather rapidly today, it is the feeling of the Guide Publication Committee that a more up to date, practical and complete chapter could

be written by such a man, acting under the direction of the group, than it could be by an individual selected by the Committee.

Although at this particular time it is early to be too elated about the success of the efforts of the Guide Publication Committee, nevertheless, it is with a great feeling of satisfaction and pleasure, and with thanks to the Sub-Chairmen of the Committee, to your Secretary, Mr. Hutchinson, to the Chairman's Assistant, Mr. Blackshaw, and to the approximately two hundred people who have assisted him with information and criticism, that this report is respectfully submitted.

W. L. FLEISHER, *Chairman*,
JOHN HOWATT,
G. L. LARSON,
S. R. LEWIS,
E. N. MCDONNELL,
W. M. SAWDON,
J. H. WALKER,
GUIDE PUBLICATION COMMITTEE.

On motion, the report of the Committee was accepted and made a part of the records.

President Haynes called the fourth session to order and later First Vice-Pres. John Howatt presided.

W. H. Driscoll, Chairman of the Committee on Ventilation Standards, reported informally for the Committee.

Immediately after calling the fifth session to order President Haynes introduced Arthur M. East, Associate Director of the Federal Housing Administration, to outline the Results of the Modernization and Building Campaign.

Vice-Pres. John Howatt took the chair and the progress report of the Committee on Code for Testing and Rating Convectors of which R. N. Trane is Chairman was read by Albert Buenger, St. Paul.

Report of Committee on Code for Testing and Rating Convectors

The work of this Committee has been quite limited this year in view of the fact that two codes have already been adopted by the Society, one for the testing and rating of convectors for steam and the second for hot water.

It has been the purpose of the Committee to cooperate in every way to make these codes of importance to the industry. We are very glad to report that the N.R.A. Code for Testing and Rating of Convectors includes a clause which states that all manufacturers must test and rate their convectors in accordance with the A.S.H.V.E. Code. This industry, through its Code Authority, has required all manufacturers to have their convectors tested and rated by the Frost Laboratories, and these ratings will be published by all manufacturers on September 1 in their catalogs and will be used as a basis for pricing.

As soon as the steam ratings have been put into use, this Committee will endeavor to have the hot water code similarly adopted.

This Committee is endeavoring to be alert to see that any recommendations for changes of the Code are promptly given to the Society. At the present time (Aug. 1934) we are considering modifying our formula, by which the conversion factor exponent of 1.3 is changed to 1.5, which has been found to be a little more accurate and very close to the average of the tests made at the University of Illinois for the purpose of substantiating the correctness of the 1.3 factor.

Supplementing the report of Aug. 20, your Committee recommends that the exponent 1.3 shown in the correction factor formula in Paragraph 27 of the Hot Water Code and in Paragraph 28 of the Steam Code be changed to 1.5.

This change is brought about as the result of a large series of tests run at the University of Illinois, indicating that the factor of 1.3 is too low and that the factor 1.5 is very close to the average obtained in the series of tests.

We are pleased to report that in the N.R.A. Code the following has been inserted:

"On and after September 1, 1934, all ratings of performance of concealed radiators shall be in accordance with the Code for Testing and Rating Concealed Radiators as established by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS."

Your Committee has cooperated with the *Concealed Heater Association* and we have been advised that all manufacturers belonging to the Association have had their units tested in an independent laboratory and that by February 1 all their convectors will be rated in accordance with the tests so obtained.

These manufacturers are rating their units by the Steam Code only, and it will be the purpose of your Committee to continue their work so that the Hot Water Code may also be adopted as a universal practice.

Respectfully submitted,

R. N. TRANE, *Chairman*,
E. H. BELING,
R. F. CONNELL,
M. DILLMAN,
W. EWALD,
J. H. HOLBROOK,
HUGO HUTZEL,
A. P. KRATZ,
M. G. STEELE,
O. G. WENDEL,

COMMITTEE ON CODE FOR TESTING AND RATING CONVECTORS.

January 14, 1935.

President Haynes opened the final session of the 41st Annual Meeting and called upon Prof. W. M. Sawdon, Chairman of the Publication Committee, to present the report of his Committee.

Report of the Publication Committee

Article B-VIII, Section 7 of the By-Laws, lays upon the Publication Committee the duty of reviewing all papers intended for publication by the Society and also approving for publication the TRANSACTIONS of the Society.

Twenty-three papers, offered for publication, have been reviewed by the Committee and eighteen of these approved. Five papers were not considered suitable for publication by the Society. All material placed in the hands of the Committee has been acted upon.

Of the eighteen papers approved ten have already appeared in the JOURNAL, four of the ten having been presented at the Summer Session at Buck Hill Falls.

No action has been taken by the Committee on material for the TRANSACTIONS.

Respectfully submitted,

W. M. SAWDON, *Chairman*,
M. C. BEMAN,
A. P. KRATZ,

PUBLICATION COMMITTEE.

January 13, 1935.

The report was accepted and made a matter of record.

Report of Meetings Committee

The Meetings Committee performed its routine functions of looking over the proposed meeting programs and advising as to their make-up. The Committee held no actual meetings, being able to transact all of its business by letter. Practically all of the work of soliciting papers and making up programs was done by the Secretary's office, the Meetings Committee acting only in an advisory and approving capacity.

At the suggestion of the Secretary, the Committee undertook to establish contact with several European members of the Society, with the view toward securing material for what might be termed a "symposium" on hot water heating or air conditioning practice. The thought was that a type problem would be presented to several European engineers for them to furnish their solution, in the form of a paper, and at the same time have an American engineer solve the problem according to

approved American practice. Due to the difficulty of conducting such a project by foreign mail, as well as to language difficulties and the inability to lend the personal urge that is so often necessary to secure contributions, the project did not reach fruition. However, contact has been established with several European engineers, and the idea introduced to them, so that another year may possibly see its accomplishment.

We believe that the coming Meetings Committee should attempt to carry this idea out.

Respectfully submitted,
W. E. STARK, *Chairman*,
F. E. GIESECKE,
L. W. MOON,
MEETINGS COMMITTEE.

President Haynes then called upon W. T. Jones, Past President of the Society, to conduct the installation of officers. The President-Elect, John Howatt, Chicago, was presented by Dr. E. Vernon Hill; First Vice-President, G. L. Larson, Madison, was introduced by W. H. Carrier; Second Vice-President, D. S. Boyden, Boston, was introduced by L. A. Harding. The members of the Council were presented by W. H. Driscoll and regret was expressed that A. J. Offner, newly elected Treasurer, could not be present because of an accident which resulted in a broken leg and his confinement to a hospital in New York.

In a brief tribute to the newly installed President, J. P. Fleming, Chicago, brought a message of greeting and good wishes from the 500 engineer custodians of the public schools of Chicago over whom Mr. Howatt has supervision.

President Howatt thanked Mr. Fleming for his message and the pledge of support.

W. A. Russell, Kansas City, presented a group of resolutions expressing the thanks of the Society to all who had contributed to the good time that had been enjoyed in Buffalo. J. J. Aeberly moved the adoption of the resolutions and after being properly seconded they were passed by a rising vote.

In closing the meeting President Howatt said that the local chapter had done a splendid job in arranging such a successful meeting and also expressed his appreciation to the authors of papers for the fine presentation of their data.

The meeting then adjourned.

PROGRAM 41ST ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL STATLER, BUFFALO, N. Y.
JANUARY 28-30, 1935

Sunday, January 27

2:30 P.M. Meeting of Council (Room 1018)

Monday, January 28

8:30 A.M. Registration—Terrace Room Foyer
10:00 A.M. First Session—Terrace Room (Lower Lobby)
Greetings by L. A. Harding, *Chairman*, Reception Committee

Response by C. V. Haynes, President
Reports of Officers

Technical Paper:
Some Recent Studies on the Ashes in Coal, Prof. T. G. Estep
Reports of Council Committees
Report of Committee on Admission and Advancement—A. J. Offner, *Chairman*
Report of Committee on Revision of Constitution, By-Laws and Rules—Thornton Lewis, *Chairman*
Report of Tellers of Election

2:30 P.M. Second Session—(Terrace Room)
Report of Committee on Research—John Howatt, *Chairman*

Technical Papers:
Cooling Requirements of Single Rooms in a Modern Office Building, F. C. Houghten, Carl Gutberlet and Albert J. Wahl
Study of Unit Room Coolers in the Research Residence, A. P. Kratz, M. K. Fahnestock and S. Konzo

6:30 P.M. Get Together Dinner—Main Dining Room—Hotel Statler

7:30 P.M. Meeting Committee on Research—(Room 1018)

8:30 P.M. Times Variety Broadcast in Terrace Room followed by Dancing in Main Dining Room

Tuesday, January 29

9:30 A.M. Third Session—(Terrace Room)
Presentation of Garage Ventilation Code—E. K. Campbell, *Chairman*

Technical Papers:
A Carbon Monoxide Alarm and Ventilation Control, F. C. Houghten and Linwood Thiessen
The Nature of Ions in Air and Their Possible Physiological Effects, Prof. L. B. Loeb
Large-Ion and the Small-Ion Content of Air in Occupied Rooms, G. R. Wait and O. W. Torreson

Report of Guide Publication Committee—W. L. Fleisher, *Chairman*

12:00 P.M. Ladies' Luncheon and Card Party—Park Lane Hotel

12:30 P.M. Luncheon Meeting of Nominating Committee—(Room 1018)

1:00 P.M. Past Presidents' Luncheon (Private Dining Room)

2:00 P.M. Fourth Session—(Terrace Room)

Technical Papers:
Sun Effect and the Design of Solar Heaters, Harold L. Alt
The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, C.-E. A. Winslow and Dr. Leonard Greenburg
A Laboratory Study of Minimum Ventilation Requirements—Ventilation Box Experiments, W. H. Lehmberg, A. D. Brandt and Kenneth Morse

Report of Ventilation Standards Committee—W. H. Driscoll, *Chairman*

7:00 P.M. Banquet and Dance—Ball Room

Wednesday, January 30

9:30 A.M. Fifth Session—(Terrace Room)
Address—Results of the National Modernization and Building Campaign by Arthur M. East, Associate Director, Federal Housing Administration, Buffalo

Technical Papers:
The Heat Requirements of Buildings, J. H. Walker and G. H. Tuttle
Tests of Three Heating Systems in an Industrial Type of Building, G. L. Larson, D. W. Nelson and John James

Report of Committee on Code for Testing and Rating Convectors—R. N. Trane, *Chairman*

2:00 P.M. Sixth Session—(Terrace Room)
Report of the Publication Committee—Prof. W. M. Sawdon, *Chairman*
Installation of Officers

Technical Paper:
Study of Summer Cooling in the Research Residence for the Summer of 1934, A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick

New Business
Resolutions

4:30 P.M. Council Meeting—(Room 1018)

COMMITTEE ON ARRANGEMENTS

M. C. Beman, *General Chairman*
Roswell Farnham, *Vice-Chairman*
W. E. Voisinet, *Banquet*
D. J. Mahoney, *Entertainment*
B. C. Candee, *Finance*
C. A. Evans, *Transportation*
P. S. Hedley, *Ladies*
Mrs. P. S. Hedley, *Ladies*
L. A. Harding, *Reception*
J. J. Landers, *Registration*
C. W. Farrar, *Publicity*

No. 1001

SOME RECENT STUDIES ON THE ASHES IN COAL

By THOMAS G. ESTEP * (*NON-MEMBER*), PITTSBURGH, PA.

DURING the last few years there has been a very rapid development in the art of burning coal for steam generation purposes, particularly in large central generating stations. During this period underfeed stokers and powdered coal systems have been brought to such a high state of perfection that it is now possible to burn efficiently almost any grade of coal at very high rates of combustion. It is very significant that this development has followed along two lines: *first*, the design of furnaces which would permit of high rates of heat release without destruction of the furnace walls, and, *second*, the design of the mechanical part of the coal burning equipment so that the ash in the coal could be disposed of continuously with little loss and that clinker trouble would be reduced to a minimum. The second part of this development required a knowledge of the composition of the ash in coal, the ash fusion temperature, the fluidity of the molten ash, the mechanism of clinker formation and, most important of all, the correlation of these factors so that if they are known, the behavior of the coal in burning can be closely predicted and suitable equipment designed.

There are so many factors entering into the subject of the behavior of ash in coal during the combustion process that field tests under actual operating conditions are of little value, because it is impossible to establish a given set of conditions and maintain them over a long enough period of time to secure accurate information. The combustion engineer, therefore, turned to the laboratory for information because in laboratory tests certain of the factors may be held constant while others are allowed to vary, and in this manner the importance of each factor may be determined and its relationship to the others established. The leaders in these laboratory studies have been the U. S. Bureau of Mines, the large coal consuming public utilities and research laboratories located in several of the engineering colleges. Valuable work has also been done in foreign countries, notably in Canada, England, and Germany.

It is the purpose of this paper to review briefly the results of these various investigations and to point out how this information may be applied advantageously to the heating industry. So far, the designers of domestic and semi-public building heating systems have looked upon ash in coal as a nuisance to be put up with and have been more concerned with its quantity rather than

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with its behavior, but, with the advent of small stokers and other modern devices for burning coal, a knowledge of ash and the part it plays in combustion becomes of increasing importance.

COMPOSITION OF ASH IN COAL

The ash in coal is usually divided into two parts, the inherent and the extraneous. The inherent ash is that incident to the plant life which went into the formation of the coal. It very rarely exceeds 4 per cent and varies within very narrow limits. The extraneous ash is the earthy material mixed with the coal forming materials during periods of inundation and during the movements of the earth's crust. This ash varies within wide limits and no attempt will be made to establish these limits. For certain studies it is desirable to attempt to separate the ash into these two divisions, but the engineer is usually interested only in the average composition of the total ash.

Coal ashes have an exceedingly complex composition made up of many constituents. The ash analysis shows only the simple chemical compounds, without indicating how these compounds may be combined in the coal. The following compounds are usually reported in an ash analysis: SiO_2, Al_2O_3, Fe_2O_3, CaO, TiO_2, MgO, P_2O_5, Na_2O, K_2O and SO_3. Usually P_2O_5 is not determined separately but is included in the Al_2O_3 and the alkalies Na_2O and K_2O are not separated.

The silicon will probably be in the form of silicates, sand and quartz; the aluminum as clay, usually combined with silicon; the iron as pyrite, marcasite, siderite, and ferrous silicates; the magnesium as carbonates and silicates; the calcium as sulphates, carbonates and silicates; the sodium and potassium as silicates, carbonates and chlorides; the inorganic sulphur as pyrite, marcasite and the phosphorus as phosphate. During the burning of the coal certain changes take place in the ash which will not be discussed here.

The various compounds reported in an ash analysis vary within wide limits as indicated by the accompanying table:

SiO_2	30 to 60 per cent	CaO	2 to 20 per cent
Al_2O_3	10 to 40 per cent	MgO	0 to 4 per cent
Fe_2O_3	5 to 30 per cent	SO_3	0 to 9 per cent
TiO_2	0 to 3 per cent	Alkalies	1 to 4 per cent

It is the great variation in ash composition which causes the wide variation in the ash fusion temperature and its behavior when the coal is burned.

ASH FUSION TEMPERATURE

The technique of ash fusion temperature determinations is fully outlined in bulletins of the *American Society for Testing Materials* and need not be discussed here. It should be mentioned, however, that the fusion temperature is always determined in a reducing atmosphere in the furnace and, as will be shown later, this gives the lowest fusion temperature obtainable with a given ash.

The ash fusion temperatures of United States coals vary from about 2000 to 3100 F.

CLINKER FORMATION

It is usually stated that coals having ash fusion temperatures below 2400 F are clinkering coals and above this figure they are non-clinkering. This is purely an arbitrary dividing line and has little bearing on the question as to whether or not a coal will produce clinker when burned. The ash fusion temperature is only one of a great many factors entering into clinker formation and it is impossible to make a prediction of how the ash will behave based on this one factor alone.

In Bureau of Mines Bulletin 209 are given the ash fusion temperatures of over 2100 United States coals. It is interesting to note that 53 per cent of these temperatures lie below 2400 F. This means that a majority of the coals in use would be classed as clinkering coals if the above arbitrary division were used.

An examination of the ash from a coal furnace will show that, although there may be no clinkers, nearly all of the ash was fused. The only exceptions to this are in cases where the ash was very highly refractory or where the combustion rate was very low. In other words, the fuel bed temperature is nearly always high enough to fuse the ash. The ash gradually gravitates to the bottom of the fuel bed and there it is acted upon by two opposed conditions, the radiation of heat downward from the burning fuel bed and the cooling effect of the combustion air coming up through the grate. If the cooling effect of the air is not sufficient to cool the ash below its fusion temperature, then clinker will result. Thus it can be seen that, while the possibility of clinker formation is reduced by using a coal of a high ash fusion temperature, it is possible, by using proper precautions, to burn coals with low ash fusion temperatures without clinker. There is no better way known to produce clinker than to close the ash pit doors with a good fire going in the furnace. This shuts off the flow of air through the grate and its cooling effect on the ash is lost. A heavy fire will produce the same result, due to the increased resistance to air flow. If the fire is stirred deeply with the firing tools the ash will be brought up from the grate into the hot fuel bed, where it will be fused again and clinkers will form.

SULPHUR IN COAL

There is a great deal of misinformation concerning the cause of clinkering. For example, many believe that a coal which is high in total sulphur will cause considerable clinker formation. This idea is wrong. Sulphur appears in coal in two forms, organic and inorganic, and the relative proportions of these two forms vary within wide limits. Organic sulphur will burn to sulphur dioxide or trioxide and has no effect whatsoever on the behavior of the ash. The only bad effects from organic sulphur are atmospheric pollution and the corrosive action of sulphur dioxide when combined with water.

The inorganic sulphur is usually in the form of iron pyrite (FeS_2). The late Dr. Parr of the University of Illinois believed that this compound first breaks up into ferrous sulphide (FeS) and free sulphur. The ferrous sulphide fuses at a temperature of 2190 F and may act as a binder for the slate and other impurities and thus produce clinker. There is no conclusive proof, however, that this action does take place in a boiler furnace.

It has also been shown that if pyrite is burned in a highly oxidizing atmos-

phere, that is, with a large excess of oxygen, the iron will be oxidized to ferric oxide (Fe_2O_3) and this will combine with the silica in the ash to form high fusion temperature compounds. If the atmosphere is slightly reducing, that is, a deficiency of oxygen, the iron becomes ferrous oxide (FeO), which will combine with silica to form readily fusible silicates. If the atmosphere is highly reducing, metallic iron will result from the pyrite. Thus there are three possibilities from the pyrite, depending upon the furnace atmosphere. Two of

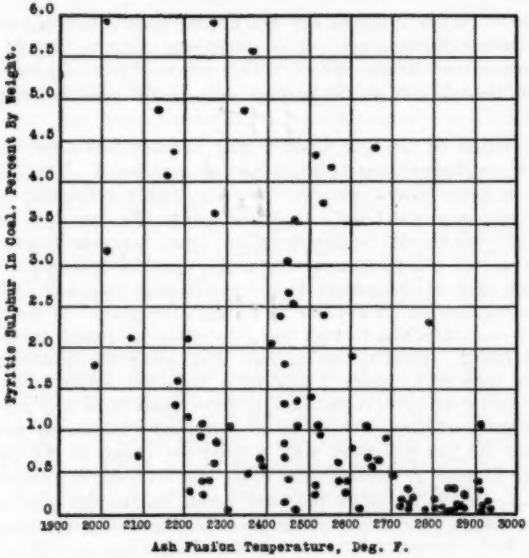

FIG. 1. PYRITIC SULPHUR AND ASH FUSION TEMPERATURE

these possibilities will result in raising the fusion temperature and one in lowering it. In a boiler furnace it is possible to have all three types of atmospheres at the same time. If the fuel bed does not offer uniform resistance to the draft, at some places there will be an excess of oxygen and in others a deficiency.

There is no doubt but that iron pyrite plays some part in clinker formation, but it is only one of many factors. At the Second International Conference on Bituminous Coal the author presented a paper on the subject of clinkering. Included in this paper was a diagram of pyritic sulphur plotted against ash fusion temperatures, showing that no relationship existed between the two. At the time this diagram was prepared, not many points were available. Since that time more information has been secured and the diagram is repeated here (Fig. 1) with more than 50 points added. These additional points only go to confirm the original statement that no definite relationship exists between pyritic sulphur and ash fusion temperature.

However, another diagram has been prepared (Fig. 2) which does indicate a little more definitely the part played by iron in coal ash when combined with silica and alumina. The heavy line across the diagram near the top shows the melting point of mixtures, in various proportions of pure silica and alumina. The ratio of silica to alumina of 34 coal ashes has been plotted along with the ash fusion temperature on this same system of coordinates. The iron content of these ashes was also known and an attempt has been made to

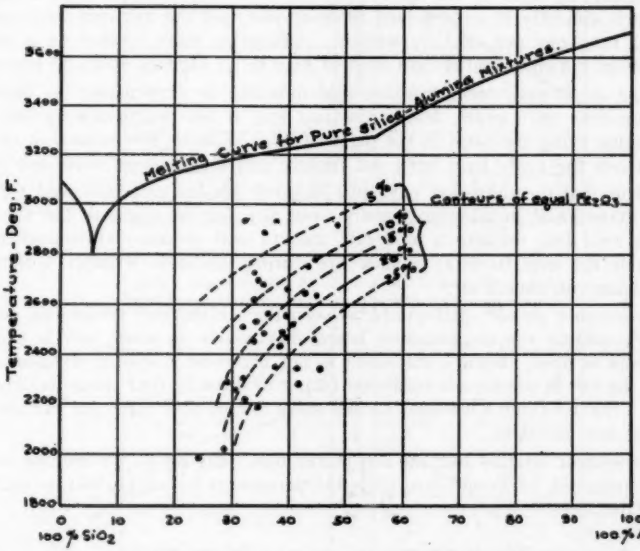

FIG. 2. THE SYSTEM $SiO_2 \cdot Al_2 \cdot O_3$. FUSION TEMPERATURES OF NATURAL COAL ASH COMPARED WITH THOSE OF SILICA-ALUMINA MIXTURES

draw contours of equal iron content as shown. There are not nearly enough points available to determine the exact position of these contours, but the general direction, at least, can be determined. There are always some points which fall outside of any general trend because of the predominance of some other ash constituent such as calcium or magnesium. For example, J. W. Greig, in the *American Journal of Science*, 1927, states that for mixtures of silica with lime or magnesia, only a very small percentage of alumina is necessary to produce a marked lowering of the fusion temperature. As a direct cause of clinkering the case against pyritic sulphur is not complete, and much more information is needed before definite conclusions can be drawn.

An investigation is in progress at the Carnegie Institute of Technology where synthetic coal ashes are being made up from pure chemicals and the fusion temperature compared with that of an identical natural ash. Not enough work has been done yet to draw any conclusions, but if it is found that synthetic ash has approximately the same characteristics as natural ash, then it

will be possible to determine the effect of each ash constituent by varying the quantity of one constituent while the others are held constant.

QUANTITY OF ASH

The impression prevails among many coal consumers that coals which are high in ash are more likely to produce clinker than those with a low ash content. The quantity of ash has no relationship whatsoever with clinker formation. For example, in some of the lumber industries where wood waste is used for fuel, much difficulty is experienced from clinker, yet the ash content is probably less than one per cent by weight. Anthracite culm is used as a steam fuel without difficulty and its ash content may be as high as 50 to 60 per cent.

Strange as it may seem, considerable difficulty is experienced in burning petroleum coke on a grate, due to the fact that it has practically no ash, the only residue being the sand in the original oil. Without the protection of the layer of ash the grate bars burn out rapidly and maintenance costs are high.

It is true that the consumer pays just as much for the ash as is paid for the coal substance and in addition usually must pay for its disposal, but the ash is not a total loss because a moderate amount will reduce maintenance costs and enable the user to carry a slow fire during moderate weather without a large amount of excess air.

The consumer should distinguish between the weight and volume of ash in coal. Frequently the complaint is heard that nearly as much ash is carried out as coal is fired. Such a statement is not necessarily a great exaggeration because the ash in some coals swells on fusing to three or four times its original volume. Some of the Clearfield cannel coals are of this type, yet the ash by weight is not excessive.

If a consumer believes that the ash is too high, judging by the volume which must be removed, he should determine the percentage by weight before making a complaint.

EFFECT OF MIXING COALS ON THE ASH FUSION TEMPERATURE OF THE MIXTURE

A very extensive study was made at the Carnegie Institute of Technology of the effect of mixing coals on the ash fusion temperature of the mixture and the results published in Bulletin 62, *Mining and Metallurgical Investigations.*

Fourteen coals from the Western Pennsylvania district having a wide variation in ash analyses and fusion temperatures were used in this study. Only two different coals were mixed at a time and usually two proportions were made for each mixture.

The ash fusion temperature of each coal was determined separately and then the ashes from two of these coals were mixed in the proportions of 60 per cent of one and 40 per cent of the other by weight and then the proportions reversed. The fusion temperatures of the two mixtures were then determined and compared with the calculated weighted average for these two ashes. To make this clear, an example will be given. Assume a mixture of coals *A* and *B*. These two coals had ash fusion temperatures of 2285 and 2665 F and ash contents of 10.2 and 7.4 per cent, respectively. A simple calculation shows that if 52.3 per cent of coal *A* is mixed with 47.7 per cent of coal *B*, the resulting

ash will be composed of 60 per cent from coal *A* and 40 per cent from coal *B*. If the ashes from the two coals are to have no action on each other, then the

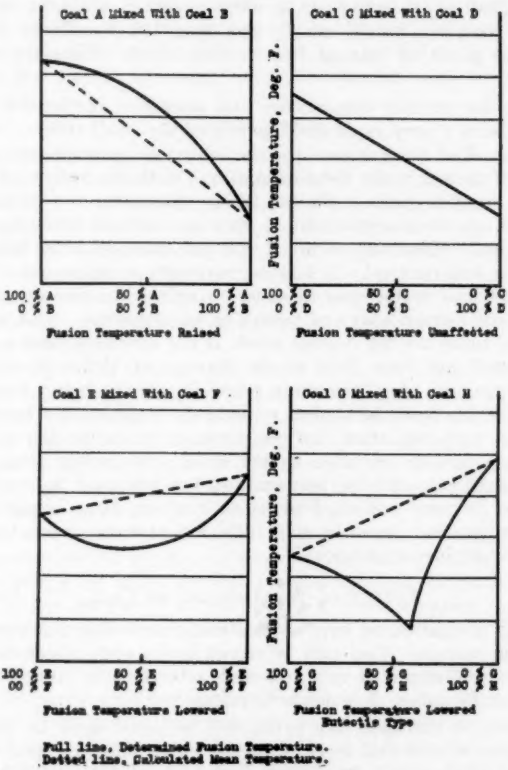

FIG. 3. EFFECT OF MIXING COALS ON THE ASH FUSION TEMPERATURE OF MIXTURE

fusion temperature of this mixture should be the calculated mean of the two. Thus,

$$\begin{aligned} &\text{For coal, } A,\ 0.60 \times 2{,}285 = 1{,}371 \\ &\text{For coal, } B,\ 0.40 \times 2{,}665 = 1{,}066 \\ &\text{Calculated mean} = 2{,}437\ \text{F} \end{aligned}$$

The actual determined value was 2495 and the result of mixing these two ashes was a slight increase in the fusion temperature.

Using the 14 coals, there were 71 different combinations of two coals each reported, and four different types of results were noted, as shown in Fig. 3.

Nine combinations showed an increase in fusion temperature, 19 no change, 43 a decrease, nine of which were of the eutectic type.

Thus about 60 per cent of the mixtures resulted in a lowering of the ash fusion temperatures; in some cases as much as 400 F, while the increase in the other 40 per cent was small, usually less than 100 F. There is no way of predicting the result of mixing two or more coals unless the ash analyses are known.

The desire for greater convenience and smokeless combustion in domestic heating has led to a very rapid development of the small stoker for bituminous coals. Nearly all of these stokers use the underfeed principle and depend upon the removal of the ash in the form of clinker. With this system of ash removal consideration must be given to the ash fusion temperature. Generally a domestic stoker will operate intermittently at very low rates of combustion and if the ash is of a highly refractory nature it will not fuse and some difficulty will be experienced in ash removal. It may be necessary in some cases to add some extraneous material to the coal in order to reduce the fusion temperature so that clinker will form readily and permit of easy removal. Just what material may be added to secure the desired result is not definitely known. Some very interesting work has been done at the Bureau of Mines in mixing fluxing agents with coal ash slag, the most promising fluxes being lime and ferric oxide, but how this could be applied to domestic fuels has not been worked out at present. A very important fact was brought out in the above investigation that there was a wide variation in the fluidity of molten slag. Two slags may have the same softening temperature, but one may be very fluid while the other may be very viscous. Fluidity or viscosity is an important factor in clinker formation, but unfortunately little experimental work has been done along this line of investigation.

EFFECT OF WEATHERING ON COAL

Weathering of coal is the term applied to the slow changes which take place in coal during storage. Coal may be stored in the open where it is subject to the atmospheric elements of rain and snow or it may be stored in a sheltered place, such as the cellar of a domestic consumer.

When stored in the open, the pyrite will be acted upon by air and water, forming sulphuric acid and ferric sulphate. The sulphuric acid will combine with some of the lime and magnesium in the ash to form sulphates. Some of the iron may go to oxides and be washed away. Weathering in the open should, therefore, raise the ash fusion temperature, and the very limited test data on this subject indicate that this is true, but more information is needed. Weathering in a sheltered place probably has little or no effect on the ash fusion temperature and very little effect on the coal itself unless spontaneous combustion takes place.

It is a prevalent custom in communities where there is no rubbish and garbage collection to use the heating plant furnace for the disposal of this refuse during the heating season. A limited study was made of this problem by Langtry and Kohout and reported in the *Proceedings of the Second International Conference on Bituminous Coal.* They found that general garbage ash had a fusion temperature of less than 2000 F but, if the glass and metal were removed, the fusion temperature was raised several hundred degrees. Various kinds of

incinerator ash were then mixed with coal ash and in all cases resulted in a marked lowering of the fusion temperature, even when added in small quantities. This investigation would seem to indicate that if trouble from clinker is experienced when burning a pure coal, it will be increased if the domestic furnace is used as an incinerator, particularly if the bottles and tin cans are not first removed. Burning of household refuse might be a partial solution to the problem of producing clinker in stoker fired domestic furnaces where clinker is desirable.

CONCLUSIONS

The subject of the ashes in coal is being studied intensively today in many laboratories. Much information has been secured, but much more remains to be done before it will be possible to determine by a coal and ash analysis just what may be expected during a given combustion process.

Some idea of the difficulties of the problem can be had by keeping in mind the complex composition of the ash itself, the variation of the amount and composition in the natural coal bed, the variation brought about by mining and preparation methods and finally by the great variation in the uses of coal and the methods of burning it.

In the final analysis the trouble experienced from coal ash will depend almost entirely on the system and method employed in burning it. Large power station operators have demonstrated their ability to burn almost all coals without difficulty and with high efficiency by studying the peculiarities of each coal and making changes in their method of managing the fires so as to avoid trouble.

To expect a high degree of skill in combustion from a domestic consumer is asking too much, and it is also impossible for a coal yard to send a skilled fireman along with every load of coal. It is believed, however, that a great deal may be accomplished by the dealer supplying each customer with printed instructions which apply to the particular coal furnished. Much of the printed matter read by the domestic consumer is of a general nature and it has its value, but he wants to know particularly how to burn the coal in his bin to the best advantage.

Once the consumer has found a coal which is suited to his equipment and which meets his needs with the minimum amount of trouble, he should try to secure this coal from season to season instead of shopping around each year and buying on a price per ton basis only. In a domestic heating plant convenience and comfort are the two most important factors and they cannot be evaluated in dollars and cents. The cheapest coal per ton is not often the cheapest coal to use.

The author believes that in the distant future gas will be the universal domestic fuel and the gas will, of necessity, be made from coal. With the proper appliances gas can be burned efficiently without skill and all of the knowledge gained from the present and future researches on coal and its ash can be applied by technically trained men to the gas making process. The domestic consumer can never be made a fuel technologist.

DISCUSSION

P. NICHOLLS[1] (WRITTEN): No comment is necessary on the data to which the author refers on the relation between fusion and ash composition. I suggest that

[1] Published by permission of the Director, U. S. Bureau of Mines.

the statements on the effect of the quantity of ash may mislead some people. It is true that, in general, with similar conditions of burning, the quality of the clinker—that is, the degree of fusion of the ash—is independent of the quantity of ash; but, on the other hand, the volume of the clinkered ash is proportional to the quantity of ash. A householder will therefore tend to have more trouble with fuels having higher ash content because the trouble increases with the size of the clinkers and their total quantity.

The foregoing statement is based on the assumption that the compositions of the ashes are the same. In practice, comparisons of coals are not as simple because increase in quantity of ash often means an increase in the quantity of shale or rock, which is refractory material; this means that the average ash will also be more refractory. However, the coal with the greater quantity of ash may still be the more troublesome because the pieces of slate may act as nuclei for the building up of clinkers, or they may be too large to pass through the openings in the grates when they are shaken.

Undoubtedly the troubles many householders experience from clinker should not occur. In our many experimental burnings of fuel in domestic furnaces it is but rarely that we have made clinkers which could be classed as troublesome. During the past three years in my home hot-water boiler (having triangular-type grates) I have burned every type and size of coke which I could find in Pittsburgh, but have never had to remove a clinker through the firing door. I always examine the ashes, and the few pieces of completely fused slag have been small. On the other hand, the coal dealers tell me that they have many complaints about clinkering from their customers using these cokes. Why is this?

The author mentions the two well-known causes of banking a hot fire and of stirring the ash into the hot fuel. He also states "that a heavy fire produces the same results, due to the increased resistance to air flow," though I do not picture what he means. We have shown [2] that, with the same rate of burning, larger clinkers are formed in a thick than in a thin bed, because below a certain thickness the temperatures through the bed are lower and also the released ash is subjected to the high temperature for a shorter time.

I suggest that the main causes of clinker trouble are failure to operate the furnace uniformly, and burning at excessive rates. Any fuels—and coke in particular—give better satisfaction if they are not rushed. Though we have not made observations in households to confirm this, it is very probable that clinkering is increased by regulators which operate the dampers on the open or completely closed principle. A little forethought and patience may prevent disagreeable labor.

It is also probable that the action of the grates when shaken affects the clinkering, and the result depends on the type of grate. I know of no published tests on grate action, but one can believe that the same type is not best suited to all fuels. The ash in the center of the bed is always more fused than toward the sides and forms a center core. If the shaking does not break up this core it will remain in the bed and be heated by the fuel lower than and surrounding it. In cold weather I always break up this core by poking through the so-called clinker door before shaking.

A highly refractory ash is troublesome because the fine dust resulting has a high resistance to air flow. An ideal ash gives a porous, friable mass, but unfortunately no ash will give the same results at all rates of burning. It must be remembered that the ash does not fuse when it is in the pieces of fuel, but only as it is released by the burning of the coke. Those interested in this subject may find some suggestive thoughts in one of the Bureau's recent publications.[3]

[2] U. S. Bureau of Mines Bulletin No. 364, p. 46.
[3] U. S. Bureau of Mines Bulletin No. 378, pp. 57-62.

No. 1002

COOLING REQUIREMENTS OF SINGLE ROOMS IN A MODERN OFFICE BUILDING

By F. C. HOUGHTEN * (*MEMBER*), CARL GUTBERLET ** AND ALBERT J. WAHL *** (*NON-MEMBERS*), PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

THE growth of the practice of cooling single segregate offices in modern office buildings has resulted in a demand for better data for estimating the required cooling capacity. A study of this subject was made by the A. S. H. V. E. Research Laboratory in the Grant Building, Pittsburgh, during the summer of 1934, in cooperation with The Technical Advisory Committee on Heat Requirements of Buildings, D. S. Boyden, *Chairman.*

Three rooms on the 27th floor (2703, 2715 and 2718, hereinafter referred to as 3, 15 and 18), having northeastern, northwestern and southwestern exposures, respectively, were chosen for the study because of their variation in exposure and because they had been used in the earlier Laboratory study [1] of heating requirements. The investigation was directed toward the development of data of value in estimating the cooling capacity needed for individual rooms in office buildings, where the demands of the occupants are limited to the period from 8:30 or 9:00 a. m. to 4:30 or 5:00 p. m.

TEST ARRANGEMENTS AND PROCEDURE

The three rooms had approximately equal outside exposures and window areas. All inside partitions, including ceiling and floor, separated them from other unconditioned rooms. Table 1 gives characteristics of the rooms which may affect their rate of heat gain.

The practice usually followed in cooling such offices is to place therein a cabinet containing direct expansion cooling coils, supplied with liquid refrigerant from an electrically driven compressor located in some adjoining space.

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** Research Asst., A. S. H. V. E. Research Laboratory.
*** Graduate Student, University of Pittsburgh.

[1] Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 387.

Presented at the 41st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by F. C. Houghten.

XUM

In this study it was necessary to make an accurate determination of the rate of heat extraction from the room by the cooling system. Since the usual cooling practice does not lend itself to such measurements, the arrangement shown in the diagrammatic sketch, Fig. 1, was used. One commercial cooling cabinet of approximately one ton capacity and one condensing unit of approximately two tons capacity were used in connection with each of the three rooms. The three combinations were loaned for the purpose by three different manufacturers of such equipment.

Each of the condensing units was connected to a copper, direct expansion cooling coil, partly immersed in water in a tank, 4 ft long by 2 ft high by

TABLE 1. ROOM CHARACTERISTICS

ROOM 3 VOLUME — 1926.8 CU. FT.						
CONSTRUCTION	RIGHT WALL	LEFT WALL	CORRIDOR WALL	OUTSIDE WALL	CEILING FLOOR	WINDOW
	C	A	A	D	E	
AREA-SQ.FT.	197.0	187.3	91.6	58.3	197.8	40.8
AIR TO AIR *	70.5	74.9	32.6	18.2	78.1	46.1
SURFACE TO AIR‡	325.0	309.0	134.6	96.2	328.4	67.3
ROOM 15 VOLUME 1069.8 CU. FT.						
CONSTRUCTION	RIGHT WALL	LEFT WALL	CORRIDOR WALL	OUTSIDE WALL	CEILING FLOOR	WINDOW
	A	A	A	D	E	
AREA-SQ.FT.	115.2	110.7	93.0	55.1	109.6	37.9
AIR TO AIR *	46.1	44.3	37.2	17.2	42.2	42.6
SURFACE TO AIR‡	190.1	182.6	153.4	90.9	180.8	62.5
ROOM 18 VOLUME = 2005.8 CU. FT.						
CONSTRUCTION	RIGHT WALL	LEFT WALL	CORRIDOR WALL	OUTSIDE WALL	CEILING FLOOR	WINDOW
	A	A	B	D	E	
AREA-SQ.FT.	187.1	182.0	86.5	61.8	204.3	40.6
AIR TO AIR *	74.8	78.8	33.7	19.3	78.7	45.9
SURFACE TO AIR‡	308.7	316.8	146.0	102.0	337.4	67.0

* HEAT TRANSFER BTU PER HOUR PER DEG. FAHR. DIFF. AIR TO AIR — ENTIRE WALL
‡ HEAT TRANSFER BTU PER HOUR PER DEG. FAHR. DIFF. SUR. TO AIR — ENTIRE WALL
A-3" TILE, ¾" PLASTER BOTH SIDES U= .400
B-4" TILE, ¾" PLASTER BOTH SIDES, ¾" MARBLE & WAY UP ON CORRIDOR WALL AS U=.361
C- ½" INSULATING BOARD ON 2X4 STUDDING U= .358
D-4" FACE BRICK, 4" COMMON BRICK, 4" TILE, ¾" PLASTER, 15 M.P.H. WIND U=.313
E-3" CONCRETE, 10" AIR SPACE, METAL LATH AND PLASTER U = .385

2 ft deep. Water was circulated from the bottom of this tank by a small motor-driven centrifugal pump through the cooling coil in the cooling cabinet, then through a calibrated liquid meter of the type used in gasoline stations, and a calibrated orifice flow meter, into a shallow pan with a perforated bottom, from which it returned to the tank by trickling down over the direct expansion cooling coil. Mercury thermometers and compound differential thermocouples, with their sensitive elements located at the middle of the partition, gave the temperature of the water supplied to and returned from the cooling cabinet and the temperature rise. This temperature rise and the quantity of water circulated for any period of time gave an accurate measure of the heat extracted from the room. Both the volume meter and the flow meter were calibrated and frequently checked by weighing the water discharge through them during a given period of time. The volume meter was found accurate to one per cent or better for most rates of water flow. With extremely low rates of water flow it was found to be less accurate, and for such conditions the

orifice flow meter gave more satisfactory results. The use of the two instruments served to give a continuous check on each other.

The best cooling results were obtained by keeping the expansion coils in the tank about one-half immersed in water and allowing the water from the perforated pan to trickle down over the upper portion of the tubes as in a Baudelot cooler. The water tank, water circulating pipes, and the exposed expansion valve and refrigerated pipe lines were all lagged with rock-wool insulation so as to conserve cooling. The water supply tank was of sufficient capacity to carry the cooling load for an hour or more at most cooling rates, making it possible to operate the compressor intermittently. With this arrangement, and allowing the formation of ice on parts of the immersed expansion coils, very constant temperature of water supply was maintained, which carried over from closing time of one test to the beginning of the test period the next day.

An observer made the rounds of the three rooms continuously throughout the test period, taking the necessary readings and making the necessary adjust-

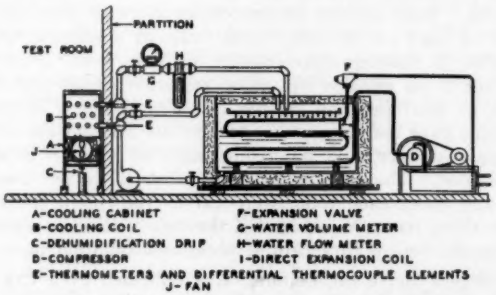

FIG. 1. DIAGRAMMATIC DRAWING OF TEST EQUIPMENT

ments. The test period began at 7:30 a. m. and closed at 4:00 p. m., Eastern Standard Time (8:30 a. m. to 5:00 p. m., Daylight Saving Time). Each room was visited at 15 or 20 min intervals. The rate of cooling within the room in order to maintain the desired temperature was manually controlled by the observer on each round by adjusting either the rate of water flow, or to some extent, the rate of air circulation through the cooling cabinet by covering part of the outlet grille with a felt pad. The water supplied to the cabinet usually ranged from 33 to 35 F, while the return water ranged from about 36 to 44 F. The control of the rate of cooling was varied between controlling the rate of water circulation and the rate of air circulation through the cabinet so as to give a water temperature differential of not less than 3 nor more than 10 deg.

The dehumidification rate was determined from the rate of water drip from the cooling cabinet. This naturally did not permit precision measurement, since a large and probably variable quantity of condensed water adhered to the cooling coils and other surfaces of the cabinet before dropping off into the collecting pan, which, having a large water surface area, permitted further

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variation in rate of water elimination compared to the simultaneous rate of condensation. However, after the system had been in operation for a few hours the rate of water elimination due to dehumidification was markedly constant.

All electrical current used within the room under test passed through a watt-hour meter which was simultaneously read with the cooling water flow data. The resulting heat supply to the room was calculated and deducted from the rate of heat extracted given by the cold water circulation measurements.

Temperatures within and about the rooms were observed as follows. The representative room temperature or control temperature was observed by a mercury thermometer in the center of the room 30 in. from the floor and shielded from both the cooling cabinet and the window. A wet-bulb thermometer located next to it with a small fan blowing air over both thermometers gave data from which the relative humidity and moisture content of the air in the room could be determined. At the same location in the room plan the temperature of the floor surface, the air temperature 6 in. above the floor, and 6 in. from the ceiling, and the ceiling surface temperature were observed by thermocouples. Wall surface thermocouples likewise gave the temperature of the surface of each of the four room walls at a point aimed to give as nearly as possible a reading representative of that wall. Such points were chosen so as not to be affected by undue radiation or air currents resulting from sunshine or the operation of the cooling cabinet. Thermocouples or thermometers also gave the temperatures of the air 3 ft outside of the window, the window glass surface, and the temperature of the air in the adjoining rooms and corridor. The constantan leads from all copper-constantan thermocouples in and about each room were carried to a cold junction. Copper leads from the three rooms were carried through a selective switch to a precision potentiometer set-up in a central observation station near room 15.

The wind velocity above the building was indicated by a cup anemometer, located at that point and connected with a counter placed on the 27th floor. The time during which each room was occupied by an observer was noted in order that the heat supplied from this source could be determined.

A mercury thermometer was placed several inches away from the masonry walls in a doorway, 5 ft above the street level, on the Third Avenue or southwestern side of the building, in order to give the air temperature in the street. Thus located, the thermometer had a *view* of the sidewalk, street and masonry buildings across the street, but not of any portion of the sky. Its reading therefore was determined by the street air temperature and radiation from the surfaces in view of it.

In order to determine the moisture content of the outside air, a wet and dry-bulb fan psychrometer was located in a corner room (2701) having a 22 ft exposure with two windows on the northeast, and a 21 ft exposure with one window on the southeast sides of the building. These windows were kept open for some time before, and during the test period. All doors leading to other portions of the building were kept closed, excepting for occasional admittance of the observer. This room was therefore subjected to free exchange of outside air, and although the temperatures indicated by the psychrometer were considerably higher on most occasions than if they had been in space outside the building, the moisture content in grains per pound of dry

air calculated from these temperatures should be quite representative of outside air.

In all but a few tests, the value chosen for the control temperature 30 in. above the floor line in the center of the room was that required by THE GUIDE standard based upon outside temperatures. For this purpose the temperature observed outside of the window on the shaded side of the building was used as the outside temperature. An effort was also made to maintain the inside moisture content specified in THE GUIDE standard. While this requirement was usually met, the dry-bulb temperature was considered the more important, and in order to maintain it without undue complication in controlling the dehumidifying load, the indoor relative humidity at times varied, usually resulting in a somewhat lower percentage.

All tests were made with the windows closed, and all but a single test were made with the doors closed, excepting for entrance and exit of the observer. All windows were equipped with inside adjustable Venetian blinds. A single test was made with the door open a part of the time in order to determine the increase in cooling load for this condition. Two tests were made, one at a relatively high temperature and one at a relatively low temperature, with several persons in the room, in order to determine the cooling load resulting from occupancy, and as a practical check on the rate of heat and moisture given off by the human body at various atmospheric conditions as determined by the A. S. H. V. E. Research Laboratory.[2] One test was made with electric lights on near the ceiling in order to determine the degree to which the resulting energy dissipation was reflected in the cooling load.

TEST RESULTS

Thirty successful tests were made during the summer on one or more of the three rooms. The calculated results are given in Table 2. The values for total heat extracted were obtained by plotting the instantaneous rates of heat extraction determined from the temperature rise and quantity of water circulated, and integrating this curve for the entire period of the test. The average rate of heat extraction from the rooms was obtained by dividing this total by the number of hours in the test. The total heat extracted for the entire test is compared with the total calculated heat gain for the same period and with the solar radiation gain.

The total calculated heat gain is taken as the sum of the transmission, solar radiation, dehumidification, and infiltration gains, and the calculated heat dissipation by the observer while in the room. The calculated transmission gain is the sum of the calculated hourly heat transfer rates from each of the walls, ceiling and floor, at their prevailing inside surface temperatures and the simultaneously determined air temperature, using a film conductance coefficient of 1.65. For the side walls, an air temperature representing the mean from ceiling to floor was used. For the ceiling and floor, the observed air temperatures 6 in. from these surfaces were used. Also in the transmission gain was included the heat transfer through the window calculated from the hourly observed air temperature 3 ft outside of the window and the window glass temperature.

[2] Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 245.

TABLE 2. ANALYSIS

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
						ROOM 3												
						HEAT EXTRACTED				CALCULATED HEAT GAIN FOR ENTIRE TEST IN 100 BTU					PERCENTAGES		HEAT EXT	
TEST NUMBER	DATE	MAX. OUTSIDE TEMP. F	MEAN OUTSIDE TEMP. F	MIN. OUTSIDE TEMP. F	TIME TEST STARTED ROOM 18	100 BTU ENTIRE TEST	100 BTU AV. PER HOUR	MAX FOR ONE HOUR: 100 BTU	MAX FOR ONE HOUR: BEGINNING HRS AFTER START OF TEST	TRANSMISSION	SOLAR RADIATION	LATENT HEAT	INFILTRATION	OBSERVER	TOTAL CALCULATED / TOTAL EXTRACTED %	SOLAR RADIATION / TOTAL EXTRACTED %	100 BTU ENTIRE TEST	100 BTU AV. PER HOUR
1*	JULY 19	93	78	62	7:30	260	30	97	0	220	23	16	8	2	103	9		
3	26	96	83	70	7:30	886	111	146	2.5	810	37	11	9	3	98	4		
4*	27	87	79	71	7:25	552	74	125	0	517	40	31			107	7		
5	30	88	75	62	7:41	631	76	117	0	564	65	7	6	4	102	10		
6*	31	81	73	65	7:40	437	58	101	2.5	425	9	17	0	3	104	2		
7*	AUG. 1	86	72	59	7:21	431	51	78	0	374	34	0	13	4	89	8		
8*	2	79	72	64	7:20	133	67	87	0	133	4	0			103	3		
9	7	87	73	59	7:38	255	33	66	0	247	27	0			107	11	215	28
10*	8	88	78	68	7:25	259	31	36	0	202	21	17	12	3	99	8	318	39
11	9	90	80	70	7:25	498	60	62	0	439	6	55	5	2	102	1	424	53
12	10	88	81	74	7:30	536	67	93	0	404	30	96	24	2	104	6	413	52
13*	13	87	76	65	7:53	315	41	63	0	239	6	63	6	1	100	2	337	43
14	14	85	78	70	7:37	358	49	69	0	252	30	77	27	2	108	8	339	44
15	15	82	74	66	7:46	409	53	90	0	400	6	68			116	1	300	39
16*	16	75	70	66	7:41	396	50	74	0	363	3	73	14	2	115	1	329	41
17*	17	85	75	65	7:53	244	33	75	0	188	10	45	16	2	107	4	230	30
18	20	74	66	58	7:35	393	48	84	0	311	46	44			102	12	369	45
20	23	79	70	62	7:34	239	31	48	0	206	5	24			96	2	158	22
21	27	78	66	55	7:33	173	23	41	0	132	29	7	8	2	103	17	136	18
22	28	70	62	53	7:44	142	19	37	4.5	141	12	2			109	8		
23	29	67	58	49	7:35													
24*	30	72	59	46	11:47													
25	31	75	64	54	7:30													
26*	SEPT. 4	77	68	58	7:35	270	35	50	1.5	243	12	23			103	4	306	37
27	5	75	66	57	7:45	166	21	39	0	151	12	4			101	7		
28	6	80	68	56	7:45	81	14	23	2.25	60	28	1			110	35	92	25
29	7	82	70	59	7:37	118	15	22	0	108	17	1			107	14	117	15
30*	10	81	70	58	7:45	216	28	45	0	191	9	20			102	4	198	26
32	12	83	74	66	8:00												277	36
33	13	74	70	66	7:45	631	87	112	3.0	431	4	186			98	1	270	37

* INSIDE VENETIAN BLIND CLOSED

The solar radiation gain when the Venetian blinds were not lowered, or when the window was bare, was taken as the total solar energy incident upon the glass surface. This was determined by computing the hourly cross-sectional area of radiation normal to the direction of the sun's rays striking the window and multiplying this by the simultaneous solar intensity normal to the direction

OF COOLING LOAD

20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39
ROOM 15									ROOM 18										
RACTED		CALCULATED HEAT GAIN FOR ENTIRE TEST IN 100 BTU					PERCENTAGES		HEAT EXTRACTED				CALCULATED HEAT GAIN FOR ENTIRE TEST IN 100 BTU					PERCENTAGES	
MAX. FOR ONE HOUR											MAX. FOR ONE HOUR								
100 BTU	BEGINNING HRS AFTER START OF TEST	TRANSMISSION	SOLAR RADIATION	LATENT HEAT	INFILTRATION	OBSERVER	TOTAL CALCULATED / TOTAL EXTRACTED %	SOLAR RADIATION / TOTAL EXTRACTED %	100 BTU ENTIRE TEST	100 BTU AV PER HOUR	100 BTU	BEGINNING HRS AFTER START OF TEST	TRANSMISSION	SOLAR RADIATION	LATENT HEAT	INFILTRATION	OBSERVER	TOTAL CALCULATED / TOTAL EXTRACTED %	SOLAR RADIATION / TOTAL EXTRACTED %
									370	42	55	0	240	78	30	16	3	99	21
									638	77	151	1.25	536	129	17	26	4	112	20
									623	86	122	4.5	536	84	40			106	13
									646	78	95	0	442	157	65	38	5	109	24
									509	66	81	1.0	367	120	24	0	3	101	24
									335	52	74	0.5	284	40	0	9	2	100	12
									132	66	73	0	130	5	0			102	4
28	65	0	188	41	0			107	19	282	36	66	0	198	157	0			126	56
39	44	0	268	10	24	15	3	101	3	385	44	57	0	248	81	32	18	3	99	21
53	63	0	315	52	55	8	3	102	12	546	69	72	7.0	377	148	73	14	2	108	26
52	69	0	327	35	56	17	2	106	8	571	69	95	0	397	140	70	20	2	110	25
43	57	0	247	27	54	7	2	100	8	432	56	67	1.0	247	112	61	6	2	99	26
44	61	0	270	59	45	11	2	114	17	425	55	77	0	281	182	44	11	2	122	43
39	59	0	231	76	44			117	25	432	56	84	0	278	172	64			119	40
41	53	0	288	24	29	5	2	106	7	447	56	77	0	332	83	68	8	1	110	19
30	37	.25	169	23	28	8	2	100	10	300	43	54	0	163	78	40	16	1	99	26
45	70	0	359	13	17			105	4											
22	34	0	139	15	9			103	9	332	42	64	0	280	106	38			128	32
18	26	0	115	33	3	6	2	117	24	324	42	67	0	223	203	1	2	2	133	63
										309	40	52	0	220	146	14			123	47
										236	28	41	0	119	231	0			148	98
										265	71	35	2.5	113	171	5			109	65
										204	28	35	6.5	81	233	7	2	2	159	114
37	54	0	286	7	27			105	2	445	54	67	0	397	61	40			112	14
										290	35	45	7.0	142	294	5			152	101
25	23	0	89	23	1			123	25	319	40	49	2.5	179	246	16			138	77
15	20	1.0	99	20	15			115	17	322	42	49	0	239	174	3			129	54
26	10	1.25	177	17	16			106	9	406	52	63	0	250	144	29			104	35
36	91	0	256	20	24			109	7	372	48	56	0	243	203	23			126	55
37	57	0	210	28	45			105	10	394	54	76	0	222	216	47			123	55

of radiation observed by the pyrheliometer. The total solar radiation impinging upon the glass at any instant is given by the formula, developed in an earlier Laboratory report.[3]

[3] Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet and J. L. Blackshaw, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 101.

$$E = P\ (\sin\ [\cos^{-1}\sqrt{\{\cos\alpha\cos(90^0-\gamma+\theta)\}^2+\sin^2\alpha}])$$
$$(wh-hL\tan(\gamma-\theta)-wL\tan\alpha+L^2\tan(\gamma-\theta)\tan\alpha) \qquad [1]$$

where:

E = solar intensity, in Btu per hour, on the window, h feet high, w feet wide, set back a distance L feet from the outside wall surface.
P = solar intensity, in Btu per square foot per hour, normal to the direction of radiation at the instant considered.
α = altitude angle of the sun at the instant considered.
γ = angle between north and the direction the window faces.
θ = azimuth angle of sun (angle between sun and the north) at the instant considered.

This paper presented justification for calculating the total heat gain through a window, by solar radiation and transmission, as the sum of the radiation incident upon the glass and the transfer (either positive or negative) from the outside air to the glass. It was again observed in this study that with intense radiation the glass temperature might be higher than both the inside and outside air temperatures, resulting in negative transfer from the outside air to the glass. This is of course accounted for by the absorption of solar radiation by the glass, resulting in a rise in glass temperature and dissipation of the absorbed energy to both the inside and outside air.

Tests made with Venetian blinds are indicated by a footnote in Table 2. For these tests the Venetian blinds were lowered and the slats set at 45 deg with the horizontal, thus excluding all direct radiation into the room through the blind. For these conditions, the sum of heat gain through the window by direct radiation and transmission calculated for a bare window was reduced to 58 per cent in making the heat balance. This is in accordance with earlier Laboratory findings [4] for a closed Venetian blind on the inside of the window.

The dehumidification load was calculated from the rate of dehumidification drippage from the cooling unit. Since considerable lag existed between the instantaneous dehumidification rate and the drippage, the average drippage rate during the later part of the test was used, and this rate was assumed to have been constant throughout the day. The infiltration rate in cubic feet per hour is given by Q in the formula

$$Q = \frac{a\ c}{b-c} \qquad [2]$$

where:

a = grains of moisture removed per hour;
b = grains of moisture per pound of dry air outside;
c = grains of moisture per pound of dry air inside;
e = cubic feet per pound of dry air 50 per cent saturated with moisture;

and the sensible heat gain by infiltration in Btu per hour is given by H in the formula

$$H = 0.018\ Q\ (t_o - t) \qquad [3]$$

where:

Q = infiltration rate in cubic feet per hour;
t_o = average surrounding outside temperature;
t = room temperature.

[4] Loc. Cit. See Note 3.

This rate was determined for the later part of the test when the dehumidification drippage rate was most constant and assumed to be constant throughout the day.

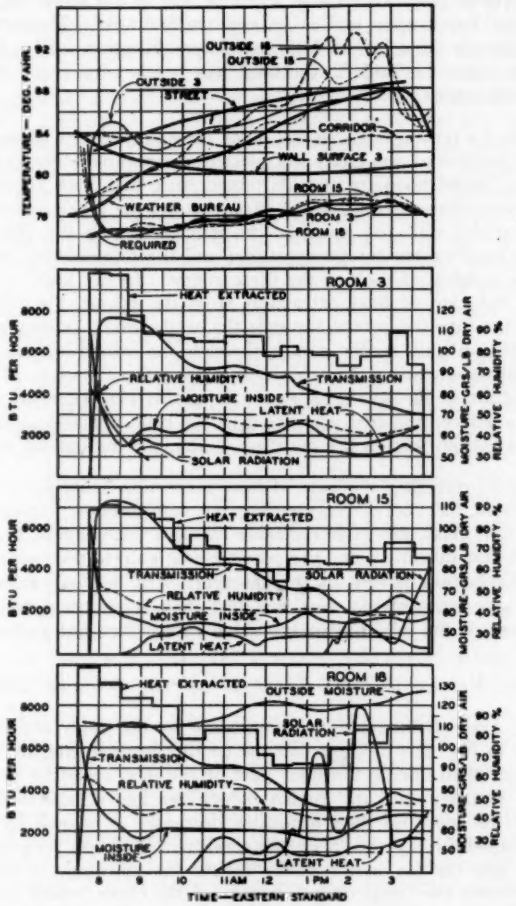

FIG. 2. RELATIONSHIPS DURING A TYPICAL TEST—NUMBER 12, AUGUST 10

In this connection it is of interest to note the magnitude of the sensible and latent cooling requirements for the air contained within the rooms for a change in conditions from 88 F and 120 grains of moisture content at the time of the beginning of the test to 77.4 F and 70 grains of moisture content. The heat

removal for this change amounted to 368, 204 and 383 Btu of sensible heat, and 1039, 577 and 1082 Btu of latent heat, for rooms, 3, 15 and 18 respectively.

The heat dissipated to the room by the observer was determined from the total of the various intervals of time when he was in the room and the rate of heat dissipation, based upon earlier Laboratory findings.[5] Theoretically, some of this had already been included in the dehumidification load. However, as shown by the values in Table 2, the total was barely significant and this correction was not made.

The results of a typical test, No. 12, made on August 9, are plotted in Fig. 2. In the upper portion of the chart are plotted the required temperature on the particular day, based upon the outside temperature and THE GUIDE standard; the control temperature of each of the three rooms; the inside surface temperature of one of the walls of room 3; the temperature in the corridor of the building; the weather bureau temperature; and the temperature observed 3 ft outside of the window of each of the three rooms. In the lower three sections of the chart the rate of heat extraction from the rooms, the calculated rate of heat transfer into the rooms through the walls, floor, ceiling and window, the solar radiation gain to the rooms through the windows, and the rate of latent heat extraction are plotted for each of the three rooms. The moisture content of the air in the room, expressed in grains per pound of dry air, and the resulting relative humidity, expressed in per cent, are also plotted for each room. In the bottom section the moisture content of the outside air is plotted.

The curves representing the rate of heat extraction from each room should equal the sum of the curves representing the transmission through the walls, floor, ceiling and glass, the solar radiation through the window, the latent heat removed by dehumidification, and the sensible heat gain to the rooms by infiltration and by dissipation of heat by the observer. The last two values are small in value and are not plotted. For this test they represented respectively 4.5 and 0.4 per cent of the total heat extracted for the entire test period.

THE EFFECT OF HEAT CAPACITY ON THE COOLING LOAD

One of the most striking points brought out by the study and demonstrated by the curves in Fig. 2 is the fact that the cooling loads during the early morning were larger than during the later part of the day, resulting from the high rate of heat transfer into the rooms, which reaches a peak soon after the beginning of the test and then gradually decreases as shown by the curves. This is accounted for by the large temperature difference between the inside wall surface and the air in the room during the morning, as shown by the difference between the temperature curves for the three rooms and the single curve, Fig. 2, giving the temperature of one wall surface in room 3. All the inside surface temperature curves plotted show this same characteristic. Due to the small heat capacity of the air within the room and the capacity of the cooling unit, the room air was brought down to the required temperature within 30 min. However, due to the high heat capacity of the walls, the wall surface temperatures did not follow this sudden drop, resulting in a maximum tempera-

[5] Loc. Cit. See Note 2.

ture difference between surface and air soon after the beginning of the test; a difference which gradually diminished until the late afternoon.

The calculated rate of heat transfer based upon the surface to air temperature difference for the left wall of room 3 during test 12 is given in Fig. 3. This rate of heat transfer is based upon the entire area of this wall, the difference in temperature between the wall surface and the room air temperature plotted in Fig. 3, and a film conductance coefficient of 1.65. The calculated heat flow for the same wall, based upon the observed air temperatures on the two sides of the partition and a transmittance coefficient, $U = 0.400$, is also plotted for comparison. It will be observed that the surface to air curve gives a much higher rate of heat transfer during the first part of the test and a much greater total heat transfer during the entire test than is indicated by the air to air curve. From the beginning of the test until 10:15 a. m. the surface to air curve gives a total heat transfer of 5145 Btu, while the air to air curve gives a transfer of 1828 Btu, or a difference of 3317 Btu. From the time of the beginning of the test until 2 p. m. the surface to air curve gives a total heat transfer of 8542, and the air to air curve, a total heat transfer of 3745, or a difference of 4797 Btu. The surface temperature curve, Fig. 2, gives a 3.5 deg surface temperature drop for the inside surface of this wall from the beginning of the test till 10:15 a. m., and 4.0 deg till 1 p. m. These temperature drops multiplied by the weight of the wall, 6855 lb or 36.6 lb per square foot, and an assumed specific heat of 0.23, give heat liberations of 5518 Btu and 6307 Btu for the two periods, assuming that the entire temperature drop applied throughout the wall; or 2759 Btu and 3153 Btu, assuming only one-half of the temperature drop to apply. The true condition will lie between these two assumptions and will entirely account for the high cooling load during the early part of the day.

Columns 9, 20 and 31 in Table 2 show a maximum rate of heat extraction usually occurring during the first hour of the test, considerably higher than the average rate. An average of all the tests tabulated for each of the rooms gives maximum rates which are 152 per cent, 162 per cent and 130 per cent of the average rate for rooms 3, 15 and 18, respectively.

The high peak in the rate of heat transfer into the room from the surrounding wall surfaces, reached during the morning hours and resulting from the large temperature difference between the required indoor temperature and the wall surface temperature, suggests the possibility of a considerable reduction in the maximum cooling load during these hours by maintaining a somewhat higher indoor cooling temperature in the morning. In order to illustrate this possibility, Test 11 on August 9, plotted in Fig. 4, was made with indoor temperatures about 3 deg higher than the low temperature dictated by the standard requirement. These temperatures are plotted in the upper portion of the chart, together with other pertinent temperatures observed during the test. The effect on the rate of heat transfer from the wall surfaces into the air and the effect on the required rate of heat extraction from the room in order to maintain these higher temperatures are also shown by the curves. The highest portion of the heat transfer curves is diminished, resulting in much flatter curves for the entire day. In fact, this change in procedure resulted in practically constant heat extraction curves for rooms 3 and 18.

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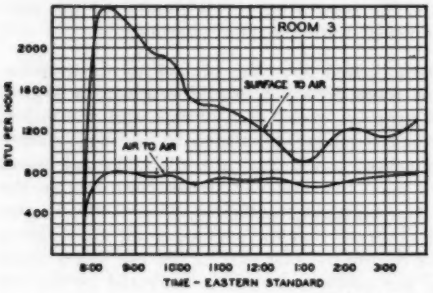

FIG. 3. CALCULATED HEAT TRANSFER BASED ON SURFACE TO AIR, AND AIR TO AIR TEMPERATURES, FOR LEFT PARTITION, ROOM 3, TEST 12, SHOWING THE EFFECT OF HEAT CAPACITY OF THE WALL

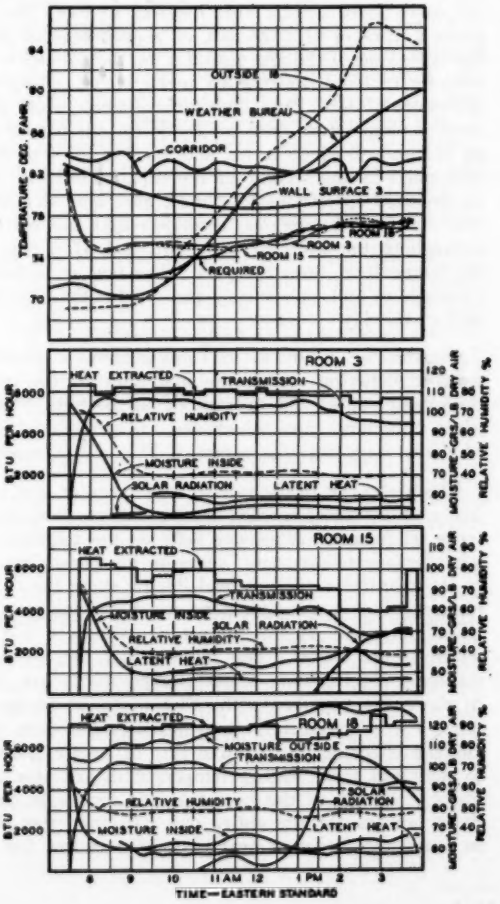

FIG. 4. RELATIONSHIPS DURING SPECIAL TEST, NUMBER 11, AUGUST 9, WHEN STANDARD REQUIRED AIR TEMPERATURE FOR COOLING WAS NOT MAINTAINED

SOLAR RADIATION THROUGH WINDOWS

Room 3 received solar radiation in the early morning, which was never great in magnitude and ceased to be effective at times ranging from 10:20 a. m. to 9:37 a. m. during the period July 1 to September 1. Room 18 began to receive solar radiation at times ranging from 10:55 to 9:55 a. m. during the period July 1 to September 1, while room 15 began to receive radiation at times ranging from 1:07 to 1:30 p. m. during the same months. On August 10, the date of Test 12, plotted in Fig. 2, there were intermittent clouds during the afternoon, diminishing considerably the heat gain from solar radiation. However, this gain as plotted for rooms 15 and 18 was sufficient to predict a much greater increase in the cooling load than given by the heat extraction curves. This was one of the surprising facts brought out by the investigation. The study of solar radiation through windows made by the Laboratory a year ago[6] indicated that radiation through a window was immediately available in the air within the room, a conclusion which is not borne out by this study. The reason for the disagreement is apparent in the fact that in the study made a year ago the solar radiation entering the room was allowed to fall on dark gray cheesecloth screens, placed between the window and floor so as to intercept all radiation and also to allow air circulation over the cloth surface. The purpose of the earlier investigation was not so much to determine the speed with which solar energy was absorbed by the air within the room, as it was to determine its rate of entrance through the window. In the present investigation no attempt was made to intercept the solar radiation by any unusual means, and as a result it fell upon the bare floor, which, in the Grant Building, consisted of a 3-in. concrete slab having cemented thereto ¼-in. imitation tile made of a rubber composition or ¼-in. brown linoleum. Since the floor had a very high heat capacity and conductivity, or a high diffusivity, the solar radiation absorbed by it was conducted downward and held in the mass, to be dissipated back to the atmosphere in both the room above and below at a slow rate over a period of 24 hours or longer. The fact that the conductance of the 3-in. concrete slab is 2.42 times greater than the film conductance coefficient from the floor surface to the air insures rapid diffusion of a larger part of the heat throughout the concrete mass. Assuming a concrete density of 150 lb per cubic foot and a specific heat of 0.23, each square foot of the floor would absorb 8½ Btu per square foot per degree temperature rise. Observations made in some of the tests, when solar intensity was high, showed this rise to be as much as 10 deg after three hours of interception of solar radiation. The area of the floor covered by the radiation through the window at any instant varied from about 30 sq ft downwards, but since the area affected did not remain stationary, but moved across the floor with change in direction of the sun, a much larger area of the floor was affected. The variable nature of the factors involved does not permit an accurate determination of the heat thus absorbed for use in making a heat balance, but it is quite probable that absorption by the floor accounted for the fact that the heat extraction curves do not rise in proportion with the solar radiation curves.

An inspection of the percentages, which the calculated heat gain by solar radiation and the entire calculated heat gain are of the total heat extracted for

[6] Loc. Cit. See Note 3.

the different tests in Table 2 when the Venetian blinds were not closed, shows satisfactory heat balances for most tests when the solar radiation gain to any room was small, but unsatisfactory heat balances when the solar radiation gain was large. The heat balance was also satisfactory in most tests when the Venetian blinds were closed, providing the heat gain by solar radiation and transfer through the window was assumed to be 58 per cent of the total calculated for the bare window. For other tests, the excess over 100 per cent which the total calculated heat gain was of the total heat extraction is approximately equal to one-half of the per cent which the calculated solar radiation gain was of the total heat extraction. As an example, in test 23, Table 2, room 18, the calculated solar radiation gain given in column 39 was 98 per cent of the total heat extracted, while the total calculated heat gain, column 38, is 148 per cent, instead of 100 per cent of the total extracted. The data indicate that particularly for rooms 15 and 18, receiving considerable solar radiation in the afternoon, only approximately half of the heat entering the unshaded windows by radiation was immediately taken up by the air, the remainder being retained

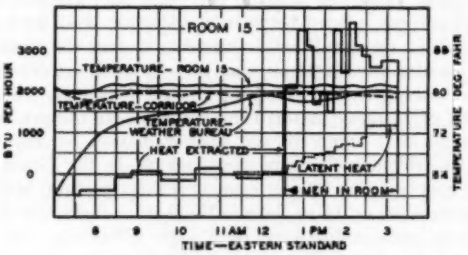

FIG. 5. EFFECT OF OCCUPANTS ON THE SENSIBLE AND LATENT HEAT COOLING LOAD

by the floor structure. The tests made with the Venetian blinds closed indicated that for this condition the 58 per cent of the calculated heat gain for a plain window estimated to enter the room did become effective in the cooling load.

It should be observed in this connection that floors of wood construction with lower conductivity and heat capacity, floors with heavy insulating rug coverings, or the presence of furniture so placed as to intercept the radiation, which was lacking in the rooms studied, would minimize the absorption of radiation and therefore the application of the results here reported. Also, that with a high rate of absorption by the floor, the lag between the solar intensity and rate of heat absorption by the air becomes a more important factor and, hence, greater solar radiation gain during the earlier part of the day may play a greater part in the maximum and total cooling load. As an example, a south window may give results which could not be predicted with sufficient accuracy from the results of this study, as was assumed in planning the test. This phase of the subject requires further study.

COOLING LOAD RESULTING FROM OCCUPANTS

Two tests were made with a number of occupants in a room in order to

determine the resulting increase in the cooling and dehumidifying loads. The results of one of these tests made on September 11 are plotted in Fig. 5. This test was made at a room temperature of 81 F, and in order to minimize the errors resulting from heat transfer into the room it was made on a day when the cooling load from sources other than the occupants was small. The cooling unit was operated at a low capacity throughout the period of the test prior to the time the occupants entered the room in order to keep it in equilibrium with the room conditions. Because the temperature maintained was high compared to the outside conditions, considerable heating had to be supplied before the men entered in order to maintain equilibrium, resulting in negative heat extraction or heating during a part of this test.

Immediately upon the entrance of the 6 men into the room, a great increase of cooling was required in order to maintain the room temperature. This increase averaged 2618 Btu per hour for the 2¾ hours of occupancy. The dehumidification rate as measured by the collection of condensation increased throughout the greater part of the period of occupancy to a constant rate of 1150 Btu per hour above the unoccupied rate. The calculated rate of sensible and latent heat dissipation from the 6 men, based upon their weight and height and the earlier findings of the Research Laboratory,[7] was 1340 and 1181 Btu per hour, respectively, or a combined rate of 2521. Considering the crudeness of the test as a calorimeter determination, the results check the laboratory data very satisfactorily.

A similar test was made at an earlier date when the entire cooling load was larger but when a lower temperature of 72 F was maintained in the room. While the results of this test did not check out quite as closely as the one plotted in Fig. 5, it also served to indicate the practical application of the Laboratory's body heat loss data, particularly, that for the lower dry-bulb temperature a larger percentage of the total heat dissipation is sensible.

EFFECT OF LIGHTS ON THE COOLING LOAD

In order to demonstrate the extent to which electrical energy used for artificial illumination, with the lights at the ceiling, increased the cooling load for individual office cooling, the test plotted in Fig. 6 was made on a day of moderate outside temperature, with a room temperature of 72 F maintained during the test period. After 4 hours of cooling, when equilibrium had been established, two electric lights of 200 w rating each and 427 w combined actual measured electric load were turned on. The combined rate at which electrical energy was supplied to the room for operating the psychrometer fan and the cooling cabinet, before the lights were turned on, and the combined load, after they were turned on, are plotted in Btu. The rate of heat extraction for the entire period of the test is plotted and shows a slowly rising cooling rate required to maintain equilibrium, which reached 1975 Btu per hour before the lights were turned on, and an average of 2320 Btu per hour for the period during which the lights were on, or an increase of 345 Btu per hour. It is probable that some of this increased cooling load resulted from increased heat gain to the room, as shown by the upward slope of the heat extracted curve before the lights were turned on. However, even without this consideration,

[7] Loc. Cit. See Note 2.

the test demonstrates that only a small part of the energy liberated by lights located at the ceiling affects the cooling load for an individual room cooled over a relatively short day. It should be pointed out, however, that this will not necessarily apply in an office building entirely cooled, particularly if the cooling is applied over a larger part of the 24 hours, since the heat stored temporarily in the ceiling and upper floor structure must eventually be accounted for. Temperature gradient curves from floor to ceiling during the period of this test, as plotted in the upper part of the chart, show clearly why the energy dissipated from the lights does not appear in the cooling load. The air heated by the lights stratifies very closely to the ceiling, raising the temperature of

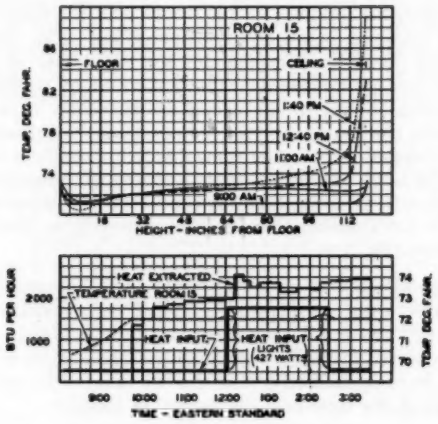

FIG. 6. EFFECT OF ARTIFICIAL ILLUMINATION ON THE COOLING LOAD

the structure, and only serves to raise the temperature of the lower layers of the air a small amount after a considerable lapse of time.

It should be observed that the 427 w for the small room, or 3.9 w per square foot of floor area used, represents a great excess over normal illumination. It was used in this case in order to magnify its measurement. Each of the two light bulbs of the Mazda type were placed in a separate fixture as shown in Fig. 7. The two fixtures were about 3 ft apart, and their closeness to the ceiling accounts for the low rate of heat absorption by the air. A low hanging fixture would no doubt give a much higher rate of heat gain to the air and thus increase the cooling load. Also a cooling unit, giving an air delivery resulting in a positive air current over the bowl and between it and the ceiling, would have the same effect.

EFFECT OF AN OPEN DOOR ON THE COOLING LOAD

Much speculation is met with in estimating cooling requirements of offices and small stores concerning the increase in cooling load resulting from frequent

opening of doors. In order to indicate the magnitude of this factor, the test plotted in Fig. 8 was made on September 12. In this test the door was kept closed until 1 p.m., when it was opened half way, or to an angle of 45 deg. The rate of heat extraction immediately increased from 5550 Btu to 10,850 Btu, or by 5300 Btu per hour. The latent heat load increased from 500 Btu per hour to 3300, and the infiltration load, calculated from the inside and outside moisture content and the dehumidification drippage, increased from 293 Btu per hour to 2140 Btu per hour. This represents an infiltration rate of 0.86 air changes per hour for the room before the door was open, and 6.2 air changes with the door open. For this test a definite temperature difference of 10 deg between the corridor into which the door was open and the room tested was maintained, this being assumed to be a representative cooling difference. The dimensions of the door were 83½ in. by 37 in. In applying the results of this test to the estimation of the effect of an open door on the cooling load of a given space, the factors which must be estimated are the difference in temperature and moisture content of the air between the cooled space and the space into which the door is opened, and the length of time per hour during which the door is assumed to be open. The resulting increase in cooling load should be independent of the wall transmission and solar radiation loads. It should be pointed out that the cooling load will not be directly proportional to the air temperature differences on the two sides of the door, but more nearly proportional to the square of this difference, since it will be directly proportional to not only the temperature difference but also to the volume of air change which in itself should increase in almost direct proportion to the temperature difference.

TEMPERATURES AFFECTING COOLING REQUIREMENTS

The inter-relationships of the temperatures plotted in Fig. 9 for the period July 15 to August 31 are of interest. The small space allowed for each day is made to represent the period from 6 a.m. to 4 p.m., Eastern Standard Time. The weather bureau temperature curves show the entire 24-hour range reported. The time of the maximum and minimum are located approximately by the upper and lower terminals of these curves. Usually, the minimum reported was around 6 a.m.; if it was earlier, the lowest end of the curve was started slightly to the left of the 6 a.m. ordinate or in the space for the previous day, and if considerably after 6 a.m. it was located slightly to the right. Likewise, the time of the maximum was indicated as coming either before or after 4 p. m. Eastern Standard Time. The shape of the curve shows approximately the rate of temperature variation during the day. The temperature of the air outside of the window of room 18 was always observed in the shade at the beginning of the period, and in the sun later, if it was a clear day. The corridor air temperature was observed on a mercury thermometer located in the corridor near room 18. The circular points give the temperature observed for the left partition of room 3 just before cooling was turned on in the morning.

The relation of the corridor temperature and its daily range and the partition surface temperature in room 3 to the weather bureau mean temperature and temperature range is of particular interest. Over a several day period of rising

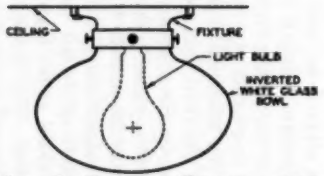

FIG. 7. ELECTRIC FIXTURE WITH INVERTED WHITE GLASS BOWL

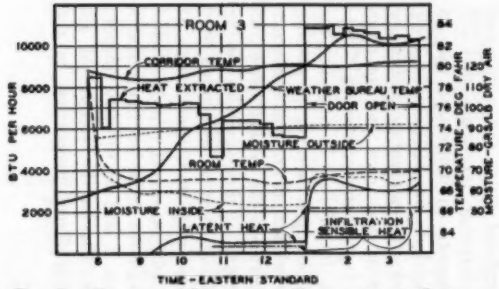

FIG. 8. EFFECT OF OPENING A DOOR ON THE COOLING LOAD

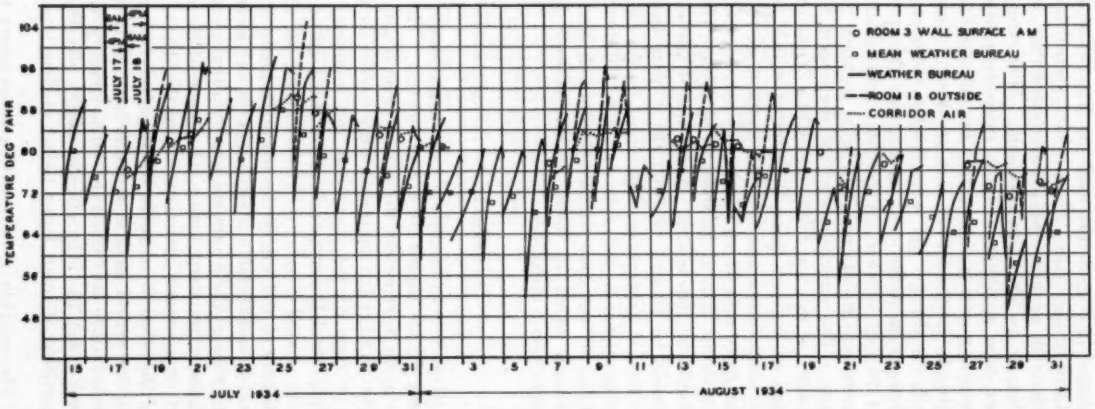

FIG. 9. VARIATION IN TEMPERATURES REPORTED BY WEATHER BUREAU, AND OBSERVED IN CORRIDOR OF BUILDING, 3 FT OUTSIDE OF WINDOW 18, AND FOR THE SURFACE OF LEFT PARTITION IN ROOM 3

temperature the corridor temperature and the partition temperature lag behind the weather bureau mean temperature, while over a period of declining weather bureau temperatures the reverse is true. For the entire period, July 15 to August 31, the corridor mean temperature and the partition temperature in the early morning average 7.4 and 6.8 deg higher respectively, than the weather bureau mean temperature. This difference must be accounted for by the dissipation of heat within the building from machinery, electric lighting, occupants, and other sources. The small daily temperature range of the air in the corridor, rarely increasing by more than 4 deg during the day demonstrates the great heat capacity of the building structure and the marked tendency of the air within the building to follow the temperature of the structure. This is further demonstrated by the close agreement among the room temperatures, the corridor air temperature and the partition surface temperature in the morning, showing that the heat extraction from a single room during a day was returned during the following 16 hours so as to raise its temperature to practically the same level as observed in the uncooled rooms. In a few instances the windows of the test rooms were left open during the night after a cooling test. This had the maximum effect of reducing the air temperature difference between the outside and the test room, as observed before test in the morning, by one quarter of the difference observed for unconditioned rooms with their windows closed.

The temperature indicated by the thermometer near the street level outside the building bore an interesting relation to the temperatures plotted in Fig. 9. Its temperature range was always considerably less than the temperature outside of the window in room 18 and that reported by the weather bureau. The low temperature in the morning was usually a little above the weather bureau mean, while the high recorded in the late afternoon was usually between the high recorded for the weather bureau and that observed outside the window in 18. The location of this thermometer as described undoubtedly had much to do with the reduced range as well as the magnitude of its readings. While it does not represent general sidewalk conditions on a hot day, it probably does so as well as any other location which might be chosen. More open spaces in the street subjected to greater air circulation and view of the sky would probably give a more extended range.

ESTIMATING COOLING REQUIREMENTS

The required indoor cooling temperature, dictated by the outside temperatures, used in most of the tests are shown by the study to result in a maximum cooling load during the first hour in the morning approximately equal to the sum of the calculated heat gain from all the inside wall, ceiling, floor and glass surfaces, for the temperature difference prevailing and the latent and sensible heat load based upon the cooling of one room volume of air. For the conditions under which the study was made, this maximum was unaffected by the higher outside temperatures and the intense solar radiation during the later part of the day, and for these conditions it is apparent that any assumption which raises the required low temperature standard during the early morning will reduce the maximum cooling load approximately in proportion to the reduced number of degrees cooling. The present standard of indoor cooling temperature based upon outside temperature is somewhat ambiguous in that the location of

the outside temperature is not more closely specified. Four rather different cooling temperature standards might be chosen in the early morning, based upon either the temperature recorded by the weather bureau, the temperature at the street level, the temperature outside the window of the office cooled, or the corridor temperature of the building. The reason for the temperature standard for indoor cooling is the feeling of contrast or shock experienced by an occupant upon entering or leaving the cooled room. Obviously, a strict consideration of this point of view would eliminate the weather bureau temperature and the temperature outside of the window of the room, and limit it to a consideration of either the street temperature or the building corridor temperature. Probably, use of the corridor temperature or the mean temperature of the building would be by far the better selection, since it determines the immediate contrast experienced by the occupant. This would have the operating advantage of a much smaller temperature range throughout the day, in fact, a range so small that the mean corridor temperature would serve satisfactorily in fixing the cooling temperature for the entire day. By requiring a somewhat higher temperature during the morning hours it would have the distinct advantage of decreasing the peak load during this time. The effect this would have on a required indoor temperature may be illustrated with reference to tests 12 and 11, plotted in Figs. 2 and 4. Basing the required temperature upon the corridor temperature of the building for these days would give an indoor temperature range of 75.7 F to 76.1 F for test 12, and 75.3 F to 76.2 for test 11, and an estimated reduction of 25 per cent in the maximum rate of heat gain by transmission during the first hour. For test 11, this consideration for outside temperature would give approximately the same conditions under which this special test was made.

In applying the accepted standards for indoor summer cooling, the need for further clarification as regards the recognition of the outside effective temperature becomes apparent. Present practice is based upon the outside dry-bulb temperature only. This is probably because the standards were built up through practical applications where the effect of moisture content of the outside atmosphere could not be easily recognized. These standards are also difficult of application, since they allow only one moisture content, fixed by a constant dew point of 57 F, for any indoor temperature. Further study will undoubtedly demonstrate the practicability of allowing some range or latitude in accepted moisture content, provided the proper effective temperature determined by the outside effective temperature is maintained, thus removing an apparently unnecessary restriction in the application of summer cooling. Great difficulty was experienced in operating the cooling units so as to maintain both the required dry- and wet-bulb temperatures. When these requirements could not be met, the required dry-bulb temperature was maintained, and the wet-bulb temperature was allowed to vary slightly, but in no case so as to give more than one degree variation in the specified effective temperature.

It is of interest to apply the data to the estimation of the maximum hourly cooling load required for the three rooms for design purposes. Since the corridor temperature of a proposed building in a given city cannot be predetermined, it is desirable to base the estimated heating requirement upon some function of the weather bureau data. Inspection of temperature ranges given in Fig. 9 shows that at the peak of a high temperature wave, the mean

weather bureau temperature tends to reach or slightly surpass the building corridor temperature. As an example, on July 21, the weather bureau mean temperature reached a higher value than the corridor temperature, and again on July 25 the weather bureau mean temperature rose to equal the lower limit of the building corridor range.

The four hot days of the entire season were: July 25, with a minimum temperature of 79 F, a maximum temperature of 96 F, and a mean temperature of 88 F; July 21, with a minimum temperature of 76 F, a maximum temperature of 97 F, and a mean temperature of 86 F; June 28, with a minimum temperature of 72 F, a maximum temperature of 95 F, and a mean temperature of 84 F; and July 26, with a minimum temperature of 70 F, a maximum temperature of 96 F, and a mean temperature of 83 F. It is of interest to note that during the past summer season in Pittsburgh there occurred 9 days in June, 9 days in July and 2 days in August, with weather bureau mean temperatures of 80 F or higher.

Following the practice of heating, it would be logical to base cooling design on the average of several maximum mean temperatures occurring over the past several years. Since such records are not easily accessible, it may be logical to base our reasoning on the highest mean temperature occurring during the past summer season, or 88 F, which has been shown to agree closely with the maximum corridor temperature for the summer. This high corridor temperature also occurred on July 25 and ranged from 88 to 91 F. Accepting 88 F as a standard on which to base summer cooling in Pittsburgh, and THE GUIDE relationship between indoor and outdoor temperatures, an indoor temperature of 77.6 F would be required for such a day, giving a 10.4 deg differential between the assumed building temperature and the air temperature maintained indoors. Multiplying the sum of the surface to air transfer coefficients for all surfaces in room 3, or 1585 Btu per hour per degree temperature difference for the entire surface, times 10.4, or the assumed temperature difference, gives 16,484 Btu heat extraction from the wall surfaces per hour. A like consideration for rooms 15 and 18 gives 10,826 and 16,796 Btu per hour, respectively.

These values are obviously high, since they are based on the assumption that the maximum temperature difference would pertain throughout the first hour, which is not the case. During the first quarter, to one-half hour, the air temperature within the room fell rapidly to the required standard during which time the sensible and latent heat content of the air in the room above the standard condition was being extracted. At the end of this period the wall surface temperature had already fallen somewhat, and continued to fall during the following hour, so that for no hour was the average temperature difference between the wall surface and the air more than ¾ of the maximum assumed. Hence, a more reasonable figure for the maximum requirement would be ¾ of the maximum values calculated above, or 12,363, 8119 and 12,597 for rooms 3, 15 and 18, respectively. These values may therefore be considered as the maximum cooling requirement for the three office rooms studied for the given type of construction and exposures, and for a building in the Pittsburgh district. Again inspecting Fig. 9 with respect to the maximum corridor temperatures, it will be observed that July 26 had about the same maximum weather bureau temperature and the same corridor temperature. However, because the minimum for that day was considerably lower, the mean was also lower than for

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July 25. Since the corridor temperatures on the 26th and 25th were about the same, the maximum cooling loads should also be the same. Data for the 25th are not available, but they are available in Table 2 for the 26th. On this day the maximum cooling load for any hour for room 3 was 14,600 Btu per hour, and for room 18, 15,100 Btu per hour. It will be noted in this connection that the maximum rate did not occur during the first hour in this test, for the reason that trouble was experienced in obtaining cold enough water. As a result, the required temperature was approached gradually over the first few hours of this test. The maximum cooling loads for the 26th as given in Table 2 are higher than our present assumed requirements for the reason that lower room temperatures based upon the outside temperature were maintained during the morning. The application of the smaller temperature range here considered would have brought the values of 14,600 for room 3 and 15,100 for room 18 down at least 20 per cent or less than the estimated values of 12,363 and 12,597, respectively. In other words, these cooling rates would have quite satisfactorily cooled the two rooms to 77.6 F on July 26. Hence, it may be proposed as a result of this study that the maximum cooling requirements for a single office entirely surrounded by masonry walls, floor and ceiling may be given by the formula:

$$H = A f_i (t_s - t_a) \qquad [4]$$

Where:

H = the maximum cooling requirement for any hour.
A = total area of wall, floor and ceiling surfaces of the room in square feet.
f_i = the inside film conductance coefficient, or 1.65 Btu per hour per square foot per degree temperature difference between the wall surface and the air.
t_a = the required indoor cooling temperature for t_s, the highest mean weather bureau temperature assumed for the locality, and also assumed to be equal to the highest wall surface temperatures in the morning before cooling is applied.

In presenting this suggested method of estimating the maximum cooling load it should be strongly emphasized that it is only to be applied to: (1) single offices located in large office buildings and cooled for approximately 8 hours per day; (2) offices entirely surrounded by masonry walls at least as heavy as plastered hollow tile; (3) to offices with windows equipped with inside Venetian blinds or other provisions which will reduce the absorption by the air of the solar radiation through the window to approximately one-half that entering a bare window; and (4) to rooms with other than southern and southeastern exposures. As pointed out, it should be further emphasized that because of the unforeseen results of the study and the resulting choice of rooms not including southern and southeastern exposures, the effect of these variables should receive further study.

SUMMARY AND CONCLUSIONS

1. The study made in the Grant Building during the past cooling season clearly indicates that with the acceptance of present standards for indoor cooling temperatures the main factor in the maximum cooling load is the high rate of heat transfer from the surface of walls, ceiling and floor to the cooled air in the room, and that the peak requirement takes place soon after the beginning of the cooling period.

2. The outside temperature basis for the present cooling temperature standards is shown to have a large bearing on the maximum cooling rate and a lesser effect on the total cooling load for the day. The acceptance of the building corridor temperature instead of the temperature outside of the building as a basis for determining the required cooling temperature for single offices is shown to have some inherent advantage in giving a more desirable indoor temperature, besides the very material advantage of reducing the peak cooling load.

3. For the conditions of construction and exposure in the Grant Building, the solar radiation load had little effect on the peak cooling load for the day. Approximately the same effect on the total cooling load was had with or without the use of inside Venetian blinds. This was the case in the Grant Building, particularly because the concrete floors absorbed and retained practically the same percentage of the solar radiation entering the window as the closed Venetian blinds would have retarded. In this connection it should be pointed out that other variables including a south or southeastern window, and floors with less heat capacity and higher conductivity, may alter these findings in other buildings. These variables should be studied further.

4. The Research Laboratory's earlier findings on sensible and latent heat dissipation from persons were shown to apply to the cooling of a room.

5. The electrical current used for artificial illumination with lights placed near the ceiling was shown to have little effect on the cooling load in an office cooled during the usual short period.

6. The increase in the cooling load resulting from an open door for a single air condition representing average practice is given.

7. A suggested method for estimating the maximum cooling load for an office, based upon the highest mean weather bureau temperature is presented for limited application to single rooms in large office buildings with masonry walls, and for rooms not having a southeastern or southern exposure.

ACKNOWLEDGMENT

Acknowledgment is due Prof. John A. Dent of the Mechanical Engineering Department of the University of Pittsburgh, for student assistance in analyzing the data. In this connection, theses based on the results of the study were presented to the University as follows: Summer Cooling in a Modern Office Building, by J. E. Helquist; and Some Factors in the Cooling of Rooms in a Modern Office Building, by A. J. Wahl. Acknowledgment is also due the Management of the Grant Building for making the rooms available for the study and for other valuable assistance in making the study a success.

DISCUSSION

E. K. CAMPBELL: I should like to call attention to the similarity between the heavy cooling load in the early morning when the temperature is reduced to the initial cooling requirements and the heavy heating-up load which occurs at the same time in an intermittently-heated building.

We have made calculations on the latter, showing that in buildings of this type of construction, with reinforced concrete and fireproof partitions, that the heating-up load, which is the heat absorbed by the interior material, will in some instances almost double the heat loss of the building. I think that is the real explanation, and I believe that Mr. Houghten had the same idea in mind.

G. L. TUVE: The stratification of air at the ceiling was spoken of. Would that not be affected by the type of fan in the unit? The unit fan, I suppose, was of such a nature that the distribution at the ceiling was not greatly disturbed, judging from the fact that there was so much stratification.

In regard to the determination of the wall surface conductance, how do you decide what surface conductance to use and what conductance factor?

JOHN HOWATT: Was there any furniture in that particular office? Were there any rugs on the floor? It seems to me that that would make a difference in the amount of heat stored up from the absorption of solar radiation.

F. C. HOUGHTEN: These rooms contained little furniture, which may be considered a fault in anticipating the needs of the study. A year ago a study of solar radiation through windows and its effect on cooling was made by the Laboratory. As a result of that study, there was developed a method of determining the solar radiation through a window into a room, which assumed that all of it became immediately available as heat in the air, to be removed as part of the cooling load. In that study, considerable care was exercised to have all radiation intercepted by cloth rather than by the floor.

We did not appreciate the important part which the absorbing cloth played in last summer's study, and as a result no particular precaution was taken to intercept the sun by furniture or anything else in this summer's study. The floors of the rooms in the current report consisted of a 3 in. concrete deck, covered by ship linoleum, or composition rubber tile. The results reported in this paper must, therefore, be considered to apply accurately only to such conditions, and other consideration must be given to the rate at which solar radiation entering the window of a cooled room is effective on the cooling load, where other types of floor or radiation absorbing materials are present. As an example, the results should be quite different for rooms where the solar radiation is absorbed by light-weight wood floors, or where furniture or draperies intercept the radiation.

Answering Professor Tuve's question concerning the stratification at the ceiling: The cold air stream from the unit did not reach and scrub the ceiling. It approached an elevation near the ceiling but did not scrub it. The lights, as shown in Fig. 7, were located very close to the ceiling, so that the air stream did not remove and bring down into the lower part of the room the heat dissipated by them.

The film conductance coefficients used in calculating the heat gain of the air were those developed by Professor Rowley at the University of Minnesota and by the Laboratory at Pittsburgh, a value of 1.56 Btu per square foot per hour per degree temperature difference.

L. A. HARDING: Mr. Houghten's results check up fairly well with the figures that are being used as approximations by some of the unit cooler builders for an intermediate office. Some of them are using as a rough check on the calculations 2½ Btu per hour per cubic foot. We find that that is what his tests show, using the average heat extraction.

J. H. VAN ALSBURG: During the past few months I have been interested in spot coolers with regard to the air stream as it leaves the unit, and believe that among the manufacturers it is generally understood that in most cases it is well if the upper strata of air is not disturbed, providing the ceiling is sufficiently high.

Most standard cabinets are designed for general conditions covering rooms of various sizes and shapes, and the angle of deflection or the angle at which the air leaves the unit should be arranged so that it will not strike the ceiling at too direct an angle, because this would accent or speed the fall of the air stream. It appears from a diagram that this cabinet is oblong and that the opening through which the air travels is also rectangular, which should be a standard practice as near as possible so the periphery of the air stream is increased in order to shorten the ultimate throw and to distribute the cool air over a wider area. If the opening is round or nearly square, the stream has a tendency to carry further and usually penetrates the upper strata at too high a velocity, which appears to be unnecessary.

No. 1003

STUDY OF UNIT ROOM COOLERS IN THE RESEARCH RESIDENCE

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URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois.

The results presented in this paper were obtained in connection with the summer cooling investigations conducted during the summers of 1932 [1] and 1933 [2] in the Research Residence, Fig. 1, at the University of Illinois, by the Engineering Experiment Station under the direction of A. C. Willard, Acting Director of the Engineering Experiment Station and head of the Department of Mechanical Engineering. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Special acknowledgment is due the National Association of Ice Industries, the Utilities Research Commission of Chicago, Ill., the General Electric Co., the Carrier Corp., and the American Blower Corp. for their cooperation. Acknowledgment is also due E. L. Broderick, Research Assistant and A. F. Hubbard, Special Research Graduate Assistant, for active participation in the detailed work of the investigation.

THE principal objects of this investigation were: (1) the determination of the method of operation and the control of the unit room coolers, and (2) the determination of the performance of the unit room coolers as measured by their ice melting capacity and by their effect on the ambient air, including temperature, humidity, and motion.

A complete description of the Research Residence in which these tests were conducted is given in two previously published papers.[1, 2]

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*** Special Research Associate in Mechanical Engineering, University of Illinois.

[1] Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 95.

[2] Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 167.

Presented at the 41st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by M. K. Fahnestock.

DESCRIPTION OF APPARATUS

Two types of commercial unit room coolers, Fig. 2, were used during these tests and their location in the living room on the first floor of the residence is shown in Fig. 3.

Unit *A* was a portable insulated ice chest, consisting of two compartments, one located above the other. The upper compartment was for ice storage, with a maximum capacity of 300 lb of ice when fully loaded, while the lower compartment formed a tank for holding the melted ice and water collected from the dehumidification of the air. The ice compartment was provided with special metal surfaces, in contact with the ice on one side and exposed to the air on the other; the side exposed to the air having extended fins which

FIG. 1. VIEW OF RESEARCH RESIDENCE IN URBANA, ILLINOIS

materially increase the heat transfer from air to ice. The warm air from the room, entering through a grilled opening in the front of the unit, Fig. 2, passed under and up the back side of the ice compartment in direct contact with the ice, and with the cold, finned, metal surfaces, into the suction inlet of the twin-motor centrifugal fans. The latter ejected the cooled air into the room from the top of the cooling unit. The water resulting from the melting of the ice, together with any condensation of moisture resulting from the dehumidification of the warm room air as it came into contact with the ice and cooling surfaces, was drained into the tank in the lower compartment.

For test purposes, this unit was mounted on a portable platform scale, which was sensitive to 0.01 lb, and both the ice meltage and dehumidification rates were determined by direct weights.

Unit *B*, Fig. 2, consisted of an insulated ice storage tank of 500 lb capacity,

located in the basement, and a cooling unit placed in the living room on the first floor, Fig. 3. The cooling unit consisted of an attractively finished cabinet, which enclosed the extended surface cooling coils, dehumidification drip-pan, and fans. The cold water was pumped from the storage tank through the cooling coils, over which the warm room air was circulated by means of a twin-motor centrifugal fan which could be operated at three different running speeds. For the tests discussed in this paper the lowest fan speed was used. The dehumidification, or moisture condensed out of the air, drained from the cooling coils into a shallow drip pan beneath the coils from which it was collected and weighed.

It should be noted that both units *A* and *B* made no provision for introducing

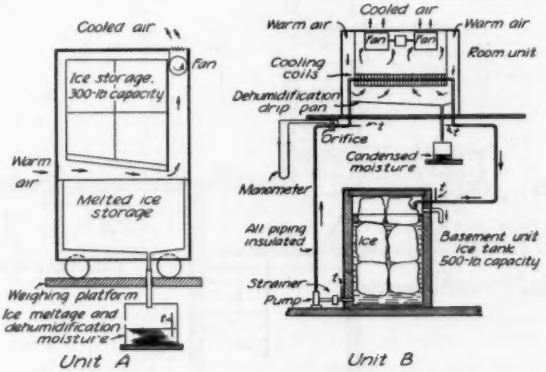

FIG. 2. DIAGRAM OF COOLING UNITS AND TEST ARRANGEMENT

air from outside of the house, but merely recirculated the air in the room which was cooled.

The first floor of the Research Residence, Fig. 3, consisted of a living room, a hallway, a dining room, a breakfast nook, and a kitchen, with a total net volume of 7300 cu ft. The living room, with a north, east, and south exposure was 13 ft 6 in. by 20 ft, with an 8 ft 11 in. ceiling, and had a total net volume of 2410 cu ft. This room contained 6 windows, having a total exposed glass area of 90 sq ft. All of the windows in the house on the east, south, and west sides, which were exposed to the sun at some time during the day, were shaded with awnings, except in the case of a few tests when the awnings were raised for the purpose of determining their effect. The average occupancy during tests was one person per room.

TEST PROCEDURE

Operating Temperatures for Room Cooling Units

One of the most important factors involved in space cooling is the debatable question of the proper inside temperature to be maintained and its relationship

to the outside temperature. Fig. 4 shows three schedules of indoor temperatures or methods of operating a unit room cooler. The schedule for the first method, designated as *Case 1*, was based on the assumption that the indoor air temperature should be maintained at a value 10 F less than the outdoor air temperature at all times. This condition is represented by the broken line, and the indoor air temperature actually maintained is shown by the solid line. It may be noted that with this schedule of operation it was necessary to start the cooler unit *A*, a considerable length of time before the maximum outdoor temperature was reached,

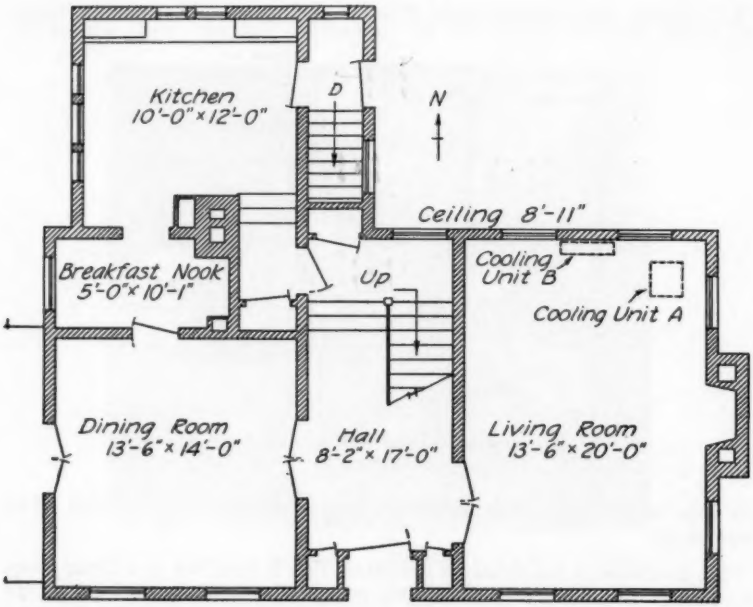

FIG. 3. FIRST FLOOR PLAN OF RESEARCH RESIDENCE SHOWING LOCATION OF COOLING UNITS

at which time the indoor temperature conditions are indicated by point *b*, and that during the period indicated between the points *b* and *c*, the outdoor air temperature increased more rapidly than the indoor air temperature. It should also be noted that after the outdoor air temperature had reached a peak value and started to decline, it was necessary to progressively reduce the indoor air temperature in order to maintain the constant differential as specified by the schedule. This reduction in indoor air temperature, accompanied by a reduction in the temperature of the contents of the room, such as furniture, inside partition walls, etc., in the late evening hours when the cooling load was just reaching

a peak due to the lag in heat transmission through the outside walls, imposed such an exceedingly heavy load on the unit as to make this schedule of operation impractical. A unit or plant of sufficient size to maintain this schedule in the late afternoon or evening would be prohibitive in first cost and uneconomical to operate. Furthermore, if the indoor conditions were conducive to comfort

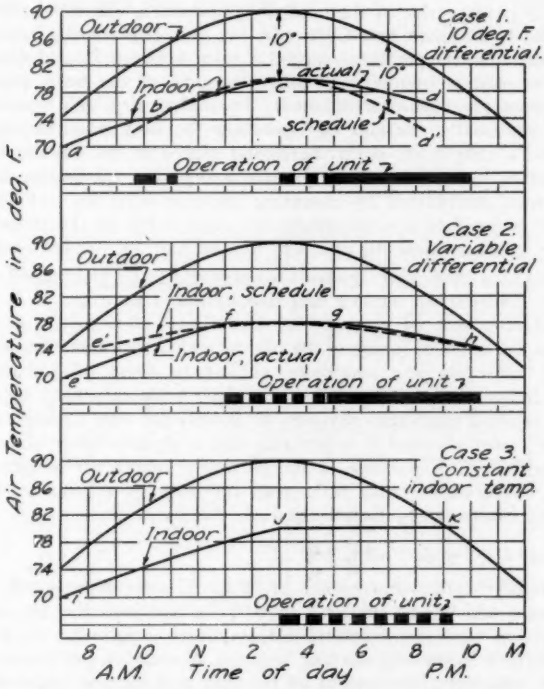

FIG. 4. THREE SCHEDULES OF ROOM AIR TEMPERATURE MAINTAINED WITH ROOM COOLING UNIT *A*

during the peak outdoor temperature, there was no indication or reason to believe that the maintenance of these conditions resulted in discomfort after the peak had passed.

In the method designated as *Case 2*, Fig. 4, an attempt was made to maintain the schedule of desirable indoor air temperatures recommended in THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' GUIDE, 1932.[3] This schedule provides for a variable differential between the indoor and outdoor temperature, the differential increasing or decreasing progressively as the outdoor temperature increases or decreases. The difficulties encountered with

[3] THE A. S. H. V. E. GUIDE, 1932, Chapter 2, p. 10, Table 2.

this method of operation were similar to those already enumerated under *Case 1*, but somewhat less pronounced due to the variable differential. In addition, both methods required careful, impractical manual control and adjustment, which would be entirely unsatisfactory in a commercial or domestic installation.

The more simple method of operation for which the cooling plant was started when the effective temperature indoors reached some predetermined value, as for example 75 F, irrespective of the outdoor temperature, was finally adopted as a standard, and is designated as *Case 3* in Fig. 4. This usually occurred when the indoor dry-bulb temperature reached a value between 78 and 82 F, depending upon the relative humidity, although in some of the tests, slightly lower dry-bulb temperatures were maintained. The indoor dry-bulb temperature was maintained practically constant by operating the unit intermittently through the action of a simple two-point thermostat placed in the electric fan circuit. This method of operation was very satisfactory, and the indoor temperature conditions were maintained by operating the unit until the outdoor effective temperature dropped to approximately the same value as the indoor effective temperature, at which time the unit was stopped and the windows were opened.

The choice of a particular dry-bulb temperature to be maintained on a given day for any domestic air cooling unit installation is governed by a number of factors, including that of economy of operation. Obviously, for economical reasons, it is advisable to maintain the highest indoor air temperature conducive to comfort. For climatic conditions similar to those of Urbana, Illinois, experience has shown that an indoor dry-bulb temperature of approximately 81 F, with relative humidities between 40 and 60 per cent is very satisfactory, whereas for cooler climates it is probable that a slightly lower air temperature might be necessary. Regardless of the particular indoor air temperature to be maintained, there seems to be little need for complex schedules of operation such as those illustrated by *Cases 1* and *2*.

Test Methods Used with Unit Cooler A

As previously stated, unit cooler *A*, Fig. 2, was mounted on a portable platform scale which was sensitive to 0.01 lb, and both the ice meltage and dehumidification rates were determined by direct weighing. At the start of each test the unit cooler was charged with ice, thoroughly pre-cooled, the water drained out, and the gross weight of the unit and ice was observed. It was then operated according to a predetermined temperature schedule, *Case 3*, Fig. 4, and at the end of each hour the ice meltage and dehumidification were drained directly into a separate container and weighed as shown in Fig. 2. At the same time, after the water had been drained out, the gross weight of the unit and ice was observed, and the net ice meltage was obtained from the difference in these hourly readings. Since the water drained from the unit each hour consisted of both the ice meltage and the moisture condensed out of the air, the net amount of moisture condensed, or the dehumidification, was determined from the difference between the total weight of water collected and the net weight of ice melted. By this method of procedure both the hourly rates of ice meltage and dehumidification were very satisfactorily obtained without handling the ice or the unit.

The majority of the tests with this unit were made with the unit located in the living room of the residence, Fig. 3, but several tests were also made

with it located in the dining room or in the hall. When located in the living room or in the dining room, the doors between that room and the hall were closed, confining the cooling entirely to the one room. When located in the hall, the doors to the living room and dining room were opened and the entire first story was cooled.

Test Methods Used with Unit Cooler B

The amount of heat absorbed by the cold water as it passed through the extended surface cooling coil in unit *B*, Fig. 2, could be determined by obtaining the weight of water circulated and the temperature rise of the water through the coil. From Fig. 5, it may be observed that although the temperatures of the entering and leaving water responded very quickly to the operation of the fan, there was sufficient lag in these temperatures after the fan was started and stopped to make it difficult to accurately determine their average values.

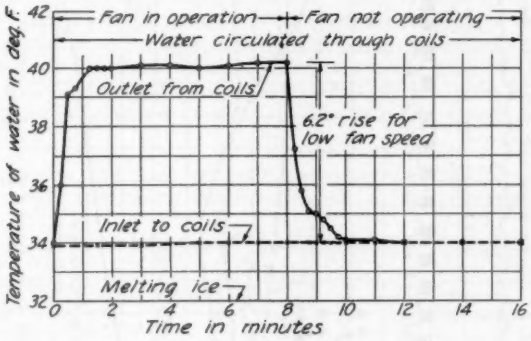

FIG. 5. TYPICAL TEMPERATURE RISE OF COOLING WATER IN UNIT *B*

Since the total temperature rise was of such small magnitude that a slight variation in either temperature resulted in a comparatively large variation in the temperature rise, and hence in the computed heat quantities, this method of measuring the heat absorption was replaced by the more direct method of weighing the ice melted during a test.

Previous to the start of a test the water in the tank and system was precooled. The weight of the initial charge of ice, and subsequent charges during a test were then accurately observed. At the end of a test the amount of ice remaining in the tank was deducted from the total amount charged, giving the net amount used in cooling and dehumidifying. As previously stated the moisture condensed on the cooling coil was drained into a container and weighed directly.

All of the tests with unit *B* were conducted with the cooling section of the unit located near the north wall of the living room, on the first floor of the Research Residence, as shown on the floor plan in Fig. 3. During some of the tests the door from the living room to the interconnecting hallway was

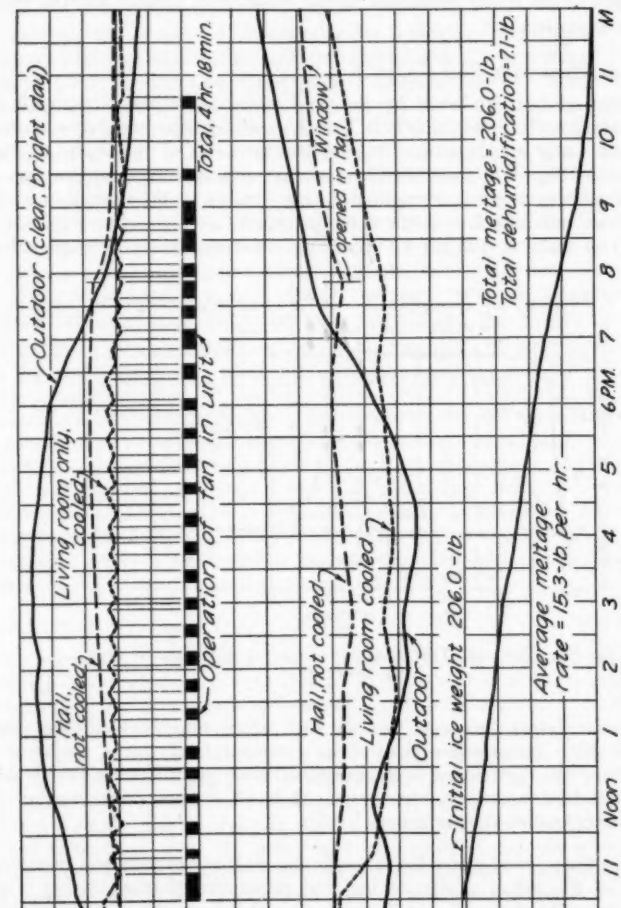

FIG. 6. TEMPERATURE AND HUMIDITY DATA FOR TEST WITH ROOM COOLING UNIT *A*. (LIVING ROOM ONLY COOLED; DOORS BETWEEN LIVING ROOM AND HALL CLOSED)

opened and the unit allowed to cool the entire first story, consisting of the living room, the hallway, the dining room, the breakfast nook and the kitchen, having a total net volume of 7300 cu ft.

As with unit *A*, unit *B* was operated on a predetermined temperature schedule, *Case 3*, Fig. 4, maintaining a fixed dry bulb temperature. It was operated on the *on* and *off* principle and was controlled by means of a room thermostat connected into the electric circuit to the fan.

TEST RESULTS

Performance of Unit Cooler A

A typical performance record for unit *A* when cooling the living room only is shown in Fig. 6. In this test the unit was initially charged with 206.0 lb of ice and the dry bulb temperature of the air in the living room was maintained at a constant value of approximately 75.5 F. The test was terminated at the end of 13.5 hours when the 206.0 lb of ice were all melted, and it was merely a coincidence that at this time the effective temperature outdoors was approximately the same as the effective temperature in the living room. It is significant to note, however, at this time, that the unit could have been stopped and cooling continued by the introduction of night air even if the ice supply had not been depleted.

The average dry bulb and wet bulb temperatures of the air entering and leaving the cooling unit during the periods of fan operation are given:

	ENTERING AIR	LEAVING AIR
Dry bulb temperature	75.0 F	63.8 F
Wet bulb temperature	64.3 F	58.0 F
Dewpoint temperature	58.6 F	54.4 F
Relative humidity	56.0 per cent	71.0 per cent
Moisture content of the air	73.57 grains per lb of dry air	62.97 grains per lb of dry air

It should be noted that the air was cooled 11.2 F in passing through the unit and that the moisture content was reduced 10.6 grains per pound of dry air.

The dry bulb temperature of the air in the living room was maintained at a level approximately 2 F lower than that in the uncooled hall due to the operation of the unit. It should also be noted that a comparable cooling effect was obtained by the reduction in the relative humidity. From Fig. 6 it may be observed that at 10:30 a. m. when the unit was started, the relative humidity of the air in the room was 68 per cent; at noon it had been lowered to 55 per cent and between that time and 8:00 p. m. it ranged from 51 to 55 per cent. In cooling effect this reduction of approximately 15 points in relative humidity is equivalent to a reduction of 1.5 deg in effective temperature or to a 2.0 F reduction in dry bulb temperature.

The total dehumidification, or the weight of moisture condensed from the air during the period when 206.0 lb of ice were melted was 7.1 lb. This change in the moisture content of the air was equivalent to a heat absorption of approximately 7474 Btu. The total heat absorbed by the ice melting process was

composed of the latent heat of fusion of the ice, plus the sensible heat absorbed by the water resulting from the meltage and was equal to

206.0 × 144 + 206.0 (57–32) = 34,814 Btu.
144 = latent heat of fusion of ice
57 F = temperature of water going to the drain.

Of the total amount of heat absorbed by the ice melting process, during the entire period of the test, approximately 21.5 per cent was given up by the change in the moisture content of the air, consisting of the total heat given up by the vapor condensed and the change in superheat of the vapor remaining in the air. As shown by the upper curve in Fig. 7, the amount of dehumidification accomplished with this type of cooling unit was dependent on the percentage of the time that the fan in the unit was operating. For example, the ratio

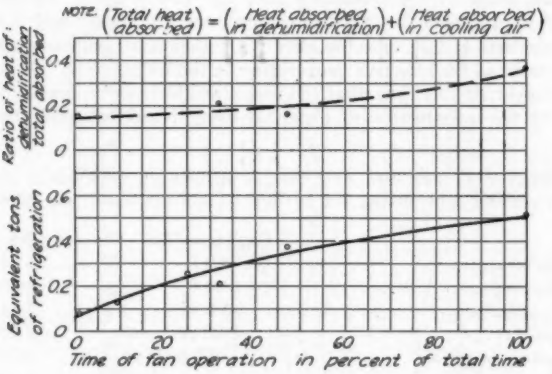

FIG. 7. PERFORMANCE CURVES FOR UNIT COOLER *A*

of the heat absorbed in the dehumidification process to the total heat absorbed, varied from a value of 37.8 per cent when the fan in the unit was operating continuously, to a value of approximately 16.0 per cent when the fan was not running and during which time the small amount of ice meltage was that due to standby heat leakage only.

The average rate of ice meltage, as indicated by the lower curve in Fig. 6, was approximately 15.3 lb per hour for a period of 13.5 hours. The maximum meltage rate observed with the fan operating continuously was 43.8 lb per hour, with a room temperature of about 75 F.

The values of heat absorption may be reduced to an equivalent ice melting capacity expressed as tons of refrigeration by dividing the heat absorption rate in Btu per hour by 12,000. Therefore, for the test shown in Fig. 6, the total heat absorption of 34,814 Btu for the 13.5 hour period was equivalent to 0.215 tons of refrigeration. The summary of results from six tests is presented

in Fig. 7 and it may be noted from the lower curve that the ice melting capacity expressed in tons of refrigeration varied from a value of 0.51 tons with continuous fan operation to a value of 0.06 tons when the fan in the unit was not running.

Performance of Unit Cooler B

The performance of unit cooler *B* was similar to that of unit *A*. However, it was possible to obtain a wider variation in cooling rates with unit *B* by making adjustments in the rate of circulation of the cooling water, by changing the speed of the fan or by allowing the fan to run a larger proportion of the time. In the tests reported in this paper the rate of circulation of the cooling water and the fan speed remained constant. The amount of cooling depended entirely upon the length of time that the fan in the unit was operated, and Fig. 5 shows the sensitivity and positiveness of this method of control as reflected in the temperature rise and fall in the cooling water, corresponding to the operation of the fan. Approximately one minute after the fan started or stopped the temperature of the water leaving the cooling coils had practically reached equilibrium. For the test shown, the temperature of the water entering and leaving the coils was approximately 34 F and 40 F respectively, giving a temperature rise of about 6 F through the coils.

Fig. 8 shows a typical performance curve with the unit located in the living room as shown in Fig. 3, and cooling the entire first story consisting of the living room, hall, dining room, breakfast nook and kitchen; or a total volume of 7300 cu ft. It may be noted that the outdoor temperature reached a maximum value of 93.7 F at about 3:30 p. m. and that the temperature in the southwest bedroom on the uncooled second floor, attained a maximum value of 87.5 F at about 7:30 p. m. During the test period of 11.67 hours it was possible to maintain average breathing level temperatures of 79.3 F in the living room, 81.5 F in the hall, 82.6 F in the dining room, and 82.1 F in the kitchen, or an average of 81.4 F for the entire first story. Although the cooling unit and control thermostat were unfavorably located with respect to the center of the first story, the air temperatures in any one room did not deviate more than 2.1 F from the mean temperature for the entire first story. The minimum room temperature of 79.3 F occurred in the living room where the cooling unit was located, while the maximum room temperature of 82.6 F occurred in the dining room. Unquestionably, this variation in air temperature between rooms would have been reduced by a more central location of the thermostat and unit. It is also probable that, from the standpoint of temperature equalization in several rooms, the best method of operation for a unit room cooler of this type is that which will allow the unit to operate for the greatest portion of the time, at a rate of cool air delivery just sufficient to balance the heat load. With such a unit equipped with a multiple speed fan this method of operation can be most closely approximated by running the fan at the lowest available speed which will handle the load; whereas, in a single speed fan unit the adjustment could be made either by employing adjustable louvers to the amount of air delivered, or by using a control valve to vary the rate of water being circulated through the cooling coils.

The total ice meltage for the test shown in Fig. 8 was 500 lb, and the total amount of heat absorbed was 82,224 Btu, which included the net rise in the sensible heat of the water in the ice tank.

The average rate of heat absorption reduced to an equivalent ice melting capacity and expressed as tons of refrigeration was 0.59 tons. In general,

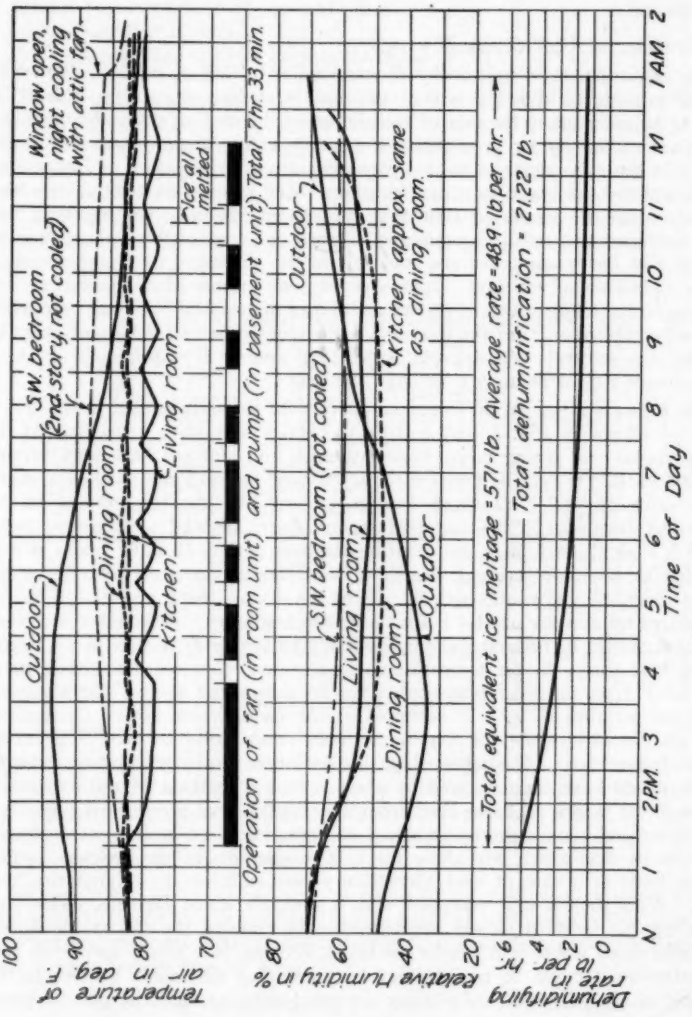
Temperature of air in deg. F.
Relative Humidity in %
Dehumidifying rate in lb per hr.
Time of Day
N 1 2PM 3 4 5 6 7 8 9 10 11 M 1AM 2
100 90 80 70
80 60 40 20
9 6 4 2 0
Window open, night cooling with attic fan
SW bedroom (2nd story, not cooled)
Outdoor
Dining room
Kitchen
Living room
Ice all melted
Operation of fan (in room unit) and pump (in basement unit) Total 7hr 33 min.
Outdoor
SW bedroom (not cooled)
Living room
Dining room
Kitchen approx. same as dining room
Outdoor
Total equivalent ice meltage = 571-lb. Average rate = 489-lb per hr.
Total dehumidification = 21.22 lb

FIG. 8. TEMPERATURE AND HUMIDITY DATA FOR TEST WITH ROOM COOLING UNIT B. (ENTIRE FIRST STORY COOLED; ALL DOORS BETWEEN ROOMS OPENED)

the tests with both units *A* and *B* on the first story indicated that a cooling unit with a capacity of approximately 0.25 ton of refrigeration per room would be required to handle the maximum cooling requirements of rooms similar to

those in which these tests were made, and in which no outdoor air other than that which occurs by natural infiltration was introduced for the purpose of ventilation. It should be noted that the windows of these rooms which were exposed to the sun were equipped with awnings, and that the occupancy load consisted of only one person per room. There was practically no power load. The estimate of 0.25 ton of refrigeration per room is based on normal running conditions, since starting conditions impose an extra load of considerable magnitude. It is very probable that a larger tonnage capacity per room would be necessary on second story rooms since the cooling load for the second story is considerably greater than that for first story and the cooled air would tend to drift downwards to the first story.

The total dehumidification or amount of moisture condensed out of the air during the entire test lasting 11.67 hours was 21.22 lb, and the ratio of the heat absorbed by the change in the moisture content of the air to the total heat absorbed was 27.2 per cent. The large magnitude of the dehumidification load is emphasized by the lower curve in Fig. 8, showing the rate of dehumidification, or the weight of water condensed out of the air in pounds per hour. It may be observed that the initial rate at the beginning of the test was approximately 5.0 lb per hour, while the average rate for the entire test period was only 1.82 lb per hour. As shown by the center group of curves in Fig. 8, this drying process substantially reduced the relative humidity of the air in all of the rooms on the first story. The average initial relative humidity of approximately 68 per cent was reduced to a value of 58 per cent at the end of one hour's operation and at the end of five hours' operation, it had attained a value of about 50 per cent. Since the dry bulb temperature remained constant the reduction in relative humidity also represented a reduction in moisture content, or the pounds of moisture present per pound of the air. The fact that during the operation of the cooling unit the moisture content on the second story remained constant indicated that the humidities in the rooms on the second story were also somewhat reduced by the dehumidification accompanying the cooling on the first story, since it was observed on other days that when the cooling unit was not operated a gradual increase in the moisture content of the air on the second story always occurred during the day.

SUMMARY AND CONCLUSION

General Summary

Although these tests indicate that a unit room cooler can be successfully adapted to cool from one to three rooms on the same floor of a residence, it is probable that the most economical method of cooling would be to operate such units in conjunction with a fan unit for the purpose of introducing cool night air into the rooms or house. It would be feasible for instance, to locate the cooling unit on the first story of a two story house and operate it during the afternoon and evening, while the second story rooms would be cooled at night by introducing night air through the open windows.

The advantageous features of portability and lower initial cost of a unit similar to unit *A* are somewhat offset by the inconvenience of having the entire unit in the living quarters. It would probably be preferable to locate the unit

in the basement and attach a short duct system from the outlet of the unit to a register on the first story, thereby avoiding the necessity for servicing the unit in the living portion of the house.

Adequate insulation of the ice storage compartment in all types of units is very essential for the minimization of standby losses. The cooling load during the summer was extremely variable with the weather conditions fluctuating rapidly, and oftentimes it was found that after the ice storage tank was filled, very little cooling was necessary due to sudden temperature changes outdoors. Under these conditions standby losses become expensive if any appreciable ice meltage takes place between operations. With units similar to type *B*, it is essential that all pipes carrying relatively cold refrigerants be thoroughly insulated.

The matter of noise of mechanical parts and of air motion is extremely important, and great care should be taken in the design and installation of any unit to reduce these features to a minimum.

Conclusions

The following conclusions are applicable to the conditions under which these tests were made:

1. Unit room coolers should be operated and controlled to maintain a constant dry bulb temperature in the room, rather than to maintain a complex schedule of indoor temperatures varying with the outdoor temperature.

2. Unit room coolers, similar to units *A* and *B*, can be successfully adapted to cool from one to three medium sized rooms on the same floor of a residence.

3. A unit cooler of 300 lb ice capacity and having a melting rate of approximately 20 lb of ice per hour would be adequate to handle the normal cooling requirements of a medium sized first story room of a residence for most summer days, if no outdoor air other than that which occurs by natural infiltration is introduced for ventilation. However, if the occupancy of the room is to be more than two persons it is essential that additional capacity be supplied for the necessary ventilation required to reduce undesirable odors.

4. The initial ice melting rate, or the melting rate during the first hour after the unit was started, was nearly twice the value of the normal melting rate.

5. Of the total amount of heat absorbed by the unit coolers, approximately from 16 to 38 per cent was absorbed in the dehumidification process.

6. In addition to the lowering of the dry bulb temperature the relative humidity of the air in the rooms was, in most cases, reduced about 15 points after three or more hours of continuous operation of the units.

7. Although not centrally located on the first story of the residence, the cooling units operated in the living room effectively cooled and dehumidified the air in the adjoining rooms and there was some indication that the reduction in relative humidity extended to the second story.

8. With unit cooler *B* located in the living room and all interconnecting doors on the first story open, the maximum deviation of the breathing level temperatures in any room on the first story from the mean temperature of the entire first story was 2.1 F.

DISCUSSION

W. M. SAWDON: How were the humidities measured, by the sling psychrometer or by some other means?

L. A. HARDING: The rate of cooling from noon until 11 p.m. was practically constant. The average rate of cooling was approximately 1.4 Btu per hour per cubic foot of space cooled, which seems to be about one-half of the amount that is ordinarily recommended by the builders of some unit air conditioners.

W. L. FLEISHER: In regard to the low requirements of these particular tests, there is no mention in the average cooling of a residence or in the average cooling of any of these rooms of the reduction in the wet-bulb of the air coming into the room from outside. This is a very important point, which influences considerably the refrigeration requirements.

I note in the test results that the average wet-bulb entering the dehumidifier, or the ice tank, was about 64 F wet-bulb; where the average or the maximum required in most guarantees in the greater number of these units is around 73 F to 78 F, an average of about 73 F in most of the cities.

It would be interesting if calculations could be made as to the difference in refrigeration effect required with a higher wet-bulb and a definite amount of infiltered air. Here we have no means of telling exactly how much air came into that room from out-of-doors. There is no calculation to show it, although it could undoubtedly be figured from the cooling effect. If that were translated into a higher wet-bulb, where the air came in, we would find how the requirements of the unit manufacturers were developed.

I feel that the conditions here do not conform to the guarantees required by the manufacturers who sell these units.

M. K. FAHNESTOCK: In regard to the first question as to how we determined the relative humidity of the air, I wish to state that the Assman type of psychrometer was used. This is an aspirating type of psychrometer, drawing air over both the wet- and dry-bulb thermometers and we used it as a means of determining wet- and dry-bulb temperatures and relative humidities.

In regard to both Mr. Harding's and Mr. Fleisher's comments we realize that the occupancy in the room was low. Probably the power load was also low. The room was located on the first story of the residence, and it may be noted that the dry-bulb temperature outside was relatively low on the days we ran the tests, but it should be remembered that they were made incidental to other tests. We believe that the conclusions, in regard to methods of operation and control, based on the performance of the units on days with moderate outside temperatures, are the same as they would have been had the tests been made on much warmer days. If only one unit was to be installed in one room on the first floor of a residence, we certainly would not recommend that a unit having a capacity of one quarter of a ton of refrigeration would cool it properly under all conditions of operation. It is well known that with such an installation there would be times when the load due to the occupants alone would exceed that amount of refrigeration.

In regard to one quarter of a ton of refrigeration per room—during tests including both the first and second floors of the residence, with from 2 to 2½ tons of refrigeration we have been able to maintain the inside air conditions at 80 F dry-bulb temperature and 50 per cent relative humidity with four occupants, when the outside temperature was 103 F. This is slightly more than one quarter of a ton of refrigeration per room.

MR. FLEISHER: It seems that the test is not really indicative of the requirements of cooling systems with the data that we now have available. One could almost get the results reported in this paper by adiabatic saturation. A low wet-bulb is

indicated, and if adiabatic saturation were properly used, in conjunction with a small amount of ice, one could do even better than that.

I feel that we are interested in results which actually require the introduction of refrigeration, and that the results that have been obtained from this test, the set-up, and everything, may be of interest from a theoretical standpoint, but they do not answer our practical problems in connection with artificial cooling of residences.

If we are going to deal with the conditions established it would be better to concentrate more on adiabatic saturation than on refrigeration.

In other words, to obtain the full value from these tests it is necessary to establish those desired conditions for which refrigeration is absolutely essential rather than those conditions which so nearly meet the adiabatic conditions, which may in some instances have a tendency to restrain people from installing the proper refrigerating equipment. I do not feel that the tests reported have been completed enough to be of great assistance to the needs of the Society.

A. P. KRATZ: With regard to Mr. Fleisher's comments, these tests were really just a fragment of some of the work that we did on the summer cooling investigation. They were conducted largely to find out what we could about the performance of the unit coolers. Mr. Fleisher is absolutely correct; we cannot interpret these tests to be representative of the total cooling conditions or total cooling load required by the house. Any interpretation in reference to that will have to be made from our tests in which we cooled the whole house with a central system.

The main object of these tests was to present a few data on the performance of the unit coolers, and we confess that our data on the unit coolers were very fragmentary.

I would like to enlarge a little on the question Mr. Harding raised as to the discrepancy between the Btu per cubic foot of air requirements that were shown by these tests and by Mr. Houghten's tests.

There are three reasons for that difference. In the first place, we had awnings on the house which, of course, cut down the cooling load as compared with what Mr. Houghten probably obtained.

Secondly, our house was not allowed to rise in temperature appreciably at night. These tests were made incidental to tests on a central cooling system, and the temperature on starting in the morning was never very much above 80 F.

The third reason is that we maintained a constant temperature of about 80 F through the whole period of the day. We made no attempt to reduce the temperature in the afternoon and thus maintain the temperature differential, and that is the time at which the large load would occur.

If we had attempted, after the peak temperatures outdoors, to bring the indoor temperature down to some temperature difference schedule, we would have found considerably greater load than we did. So, I think it is these three differences in the method of operation which can account for the difference in the cooling requirements that we found.

MR. FLEISHER: The remark was made that this was an attempt to find out the requirements of the unit cooler manufacturers in connection with household cooling. Most of the unit manufacturers are using a refrigerant, and the refrigerant temperature in almost every instance is higher than the temperature of the water which was used in this particular test, where the entering water in the unit was 34 F and the leaving water was 40 F, or an average of about 37 F.

This is considerably cooler than the refrigerant utilized by most of the manufacturers, and that would have a great difference on the differential between latent heat removed and sensible heat removed, so that probably with that low temperature

there would be a tendency to remove more latent heat than with the average temperatures used by the unit manufacturers.

When continuing the test, I feel that for any particular practical use the temperatures of the medium, either by mixing or by some other method, should be raised to a higher temperature to conform more nearly with average practice.

In that way I think you would get a decidedly different condition, both sensible temperature maintained and latent heat removed, and it would give more of an idea of the requirements of the unit system than are obtained from a test of this kind where temperatures lower than the average are used in the unit and where the leaving temperature of the water, with a 6 F rise is probably lower than the average temperature of the water delivered by the average refrigerating unit.

PROFESSOR KRATZ: The only answer I can give is that these temperatures were characteristic of the two ice units used, and the results can only be interpreted in the light of the units used, which were ice units.

MR. FLEISHER: I do not agree with Professor Kratz on that. The units themselves are probably standard units, and the medium put through them has a decided effect on the operation of those particular units, with relation to the temperature. Consequently, even though ice is used, there is no reason why the temperature, having such a marked effect on the removal of latent heat, should not be raised. It can be raised just as well with ice as with any other medium.

W. D. JORDAN: I think the reconciliation of the large space and the small capacity of the unit is found in Fig. 6. You will find on Fig. 6 that at 5 p.m., while it was 92 F out-of-doors, the hall, which was not cooled, had a temperature of 79 F, and the living-room, which was cooled, had a temperature of 76 F. So we have an actual temperature drop or differential of only 3 deg between the cooled room and the uncooled room.

In cooling a commercial space, or an office building such as Mr. Harding spoke of, when figuring on a 10 F differential between the cooled space and the corridor space, which would correspond to the hall in the house, the work done is about three times the amount of cooling effected in the residence under discussion. And, ⅓ would about coincide with the calculation of 1 Btu per cubic foot per hour in the residence as against the 3½ that we are usually called upon to furnish in commercial applications.

MR. HARDING: But the partition is approximately only ⅓ of the total wall surface.

MR. JORDAN: The point is that there is really a 3 F differential between cooled and uncooled space in the residence as against a 10 F in a commercial space, between corridor and cooled space. For some reason the temperature in the halls did not show much rise in this case. While the house is not insulated, I noticed all the way through the test that the temperature indoors, where it was not cooled, did not seem to rise very rapidly, and I think that explains why the room seemed properly cooled with a small capacity unit.

Mr. Fleisher's comment on the low temperature of the cooler as compared with a direct expansion unit having a refrigerant temperature of 41 F to 45 F, would make quite a difference in the dehumidification. Most unit cooler manufacturers are designing for 41 F to 45 F evaporator temperature which, with a proper volume of air, will maintain a 50 per cent relative humidity in the space cooled.

Our experience with unit coolers has indicated that an average room needed a minimum of ½ to ¾ ton capacity, or an ice-melting capacity of 40 to 60 lb an hour as a minimum for a unit to cool a room of the size under consideration.

C. L. TOONDER: We have been taught in practicing air conditioning for comfort that it is not only the temperature, but also the moisture content of the air that we are interested in, and that even when the temperature is sufficiently cooled to absorb

the heat losses of a body by radiation and convection the relative humidity, or the drying power of the air, must also be sufficiently low to absorb the evaporation from the body.

It is true that we lose our body heat in larger degree by evaporation than by the difference in sensible temperature. Hence, moisture content of the air surrounding the body is the most important factor. I notice from this paper that in Fig. 4, which shows the determination of the conditions under which the tests are to be made, no reference is made to moisture content or humidity. All reference is to temperature, which I think is the least important of the factors governing comfort. Thus, in order to get a paper that can be used to any practical purpose in this profession, we must have tests and results which will give us both the temperature and humidity control and conditions.

There is just one other comment in regard to the previous paper, as well as this paper: there are small differentials between the adjoining rooms and there is a considerable lag in transmission. There is a lag in the office type of building which you do not find in residences. This lag is a direct result of thermal capacity and thermal capacity is nothing more than weight times specific heat.

It is recognized that the thermal capacity in the walls of residences is not found in office building walls. Therefore, we set up a different standard of estimating for offices and a different standard of estimating for residence work.

JOHN EVERETTS, JR.: The comments made in the past few months at different meetings inferred that the latent heat load amounted to more than the sensible heat load. I can not quite appreciate that fact, inasmuch as the Society has certain definite charts and data showing that the sensible heat load is greater than the latent heat load. I should like to have that clarified.

PROFESSOR KRATZ: I might cite, in reference to the 80 F temperature, that ostensibly we were concerning ourselves only with the dry-bulb temperature. Actually the 80 F temperature that we chose for our comfort condition was based on the observation that, after the machine had been operating, a relative humidity of approximately 45 to 50 per cent was obtained. With 80 F dry-bulb and 45 to 50 per cent relative humidity, that represented, according to my recollection, about 74.5 effective temperature, which is just on the upper edge of the comfort chart.

No. 1004

A CARBON MONOXIDE ALARM AND VENTILATION CONTROL

By F. C. HOUGHTEN* (*MEMBER*) AND LINWOOD THIESSEN ** (*NON-MEMBER*)
PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

CARBON monoxide poisoning is recognized as one of the four major industrial hazards. It is met with in the home, resulting from imperfect combustion of natural gas, oil, or solid fuels, accompanied by improper elimination of the products of combustion. Casualties frequently result from exhaust gases of automobiles whose engines are allowed to run in single car garages with closed doors. Workmen in large commercial garages and other industrial establishments without proper ventilation, frequently suffer headaches and other symptoms, while less frequently they must be removed in an unconscious condition for hospital treatment.

The ventilation of commercial garages and vehicular tunnels to eliminate this hazard has received a great deal of attention during the past several years. A committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in developing a code for the ventilation of garages was confronted with a need for more technical data concerning the production of carbon monoxide by automobiles, and satisfactory methods for and the amount of ventilation required to insure satisfactory conditions at a minimum cost. Studies made by the Research Laboratory [1, 2, 3, 4] have supplied much of this information. A very great need has persisted for a satisfactory and reliable instrument at a reasonable cost to either indicate the carbon monoxide concentration in the space to be ventilated, or to be used for automatically controlling the air supply

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[1] Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 425.

[2] Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 439.

[3] Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 395.

[4] Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 263.

Presented at the 41st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by F. C. Houghten.

in order to hold the carbon monoxide concentration below the limits of safety.

The pyrotannic acid method [5] and the iodine pentoxide method [6] of analyzing samples of air for carbon monoxide have been available for laboratory purposes, but they are of no value to the operator of a garage. The Bureau of Mines

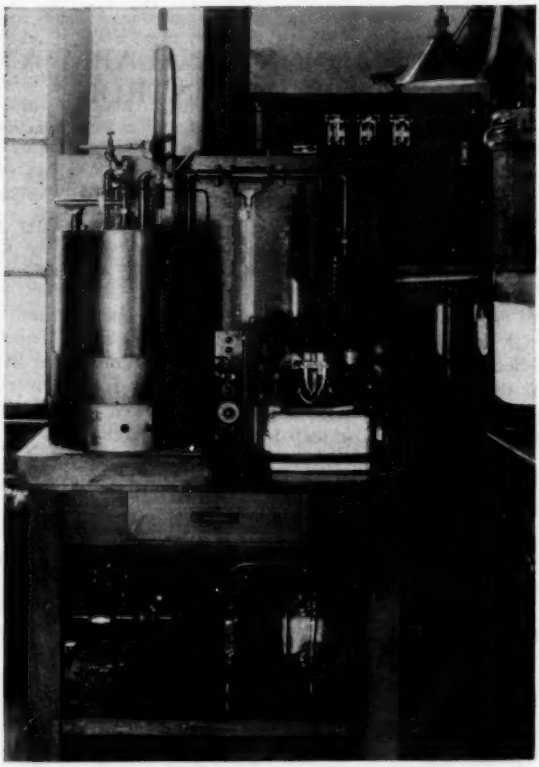

FIG. 1. U. S. BUREAU OF MINES CARBON MONOXIDE RECORDER

carbon monoxide recorder,[7] developed by Katz and others, is admirably suited for this purpose. It has, however, a much higher sensitivity than necessary,

[5] The Pyrotannic Acid Method for the Quantitative Determination of Carbon Monoxide in Blood and Air, by R. R. Sayers, W. P. Yant and G. W. Jones. Report of Investigations, Serial No. 2486, Bureau of Mines, June, 1923, 6 pp.; Technical Paper 373, Bureau of Mines, 1925, 18 pp.

[6] The Determination of Carbon Monoxide in Air Contaminated with Motor Exhaust, by M. C. Teague, *Journal Industrial and Engineering Chemistry*, Vol. 12, 1920, p. 964.

[7] U. S. Bureau of Mines Technical Paper, 355, by Katz, Reynolds, Frevert and Bloomfield (1926).

and its cost is beyond the reach of most applications. It has been successfully used in large establishments, such as the Holland Tunnel.

Largely as a result of the need expressed by the Society's Committee on Ventilation of Garages, the Mine Safety Appliances Co. of Pittsburgh, manufacturers of the Bureau of Mines carbon monoxide recorder, was induced to undertake the development of a less but sufficiently sensitive and much less expensive instrument based upon the same principle. This effort has finally resulted in the instrument here reported, which is believed to be sufficiently sensitive and reliable for the purpose.

The principle of measuring the carbon monoxide concentration of the air

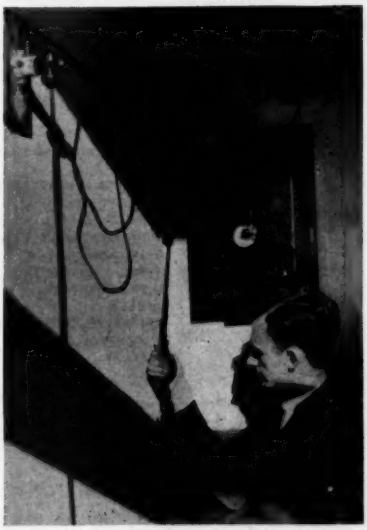

FIG. 2. THE CARBON MONOXIDE ALARM AND VENTILATION CONTROL IN OPERATION WITH PROVISION FOR SAMPLING THE ENTERING AIR

made use of by Katz and others in the Bureau of Mines carbon monoxide recorder is based upon the rise in temperature of air while passing through an oxidizing catalyst which oxidizes the carbon monoxide contained in the air to carbon dioxide with the liberation of heat. The resulting rise in temperature is indicated and recorded as an emf by a compound thermocouple with alternate junctions so placed that the air passes through one set before and through the other set after passing through the catalyst. In order to insure the extreme sensitivity desired in the Bureau of Mines carbon monoxide recorder, it was necessary to resort to a complicated system of purifying the air of its moisture content, dust and other oxidizable compounds, including partially burned gasoline and other hydrocarbons, without affecting its carbon monoxide content

before it passed through the active cell. Fig. 1 is a photograph of the Bureau of Mines carbon monoxide recorder. The oxidizing catalytic agent used in the cell is known by the trade name of Hopcalite, and was developed at the United States Bureau of Mines during the World War as an important reagent for use in the canisters of gas masks.

A photograph of the carbon monoxide alarm and control as used in this study is shown in Fig. 2. A diagrammatic drawing of the instrument is shown in Fig. 3. Air from the space to be ventilated is drawn through the dust filter, *A*, by the fan, *B*. It then passes over the electric heater, *C*, and divides through the two cells, *D*, and *E*, one containing a granular material activated with Hopcalite, and the other containing an inactive granular material of the same resistance to the flow of air and of the same heat capacity. A compound

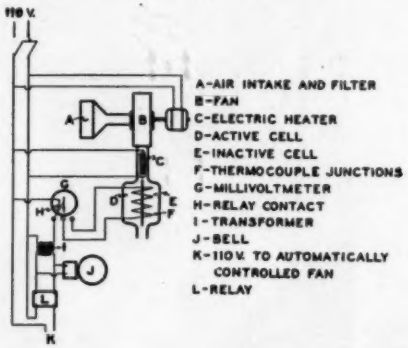

FIG. 3. DIAGRAMMATIC SKETCH OF CARBON MONOXIDE ALARM

thermocouple with alternate junctions, *F*, in the two cells sets up an emf depending on the temperature rise in the active cell. The emf resulting from the rise in temperature due to the oxidation of the carbon monoxide in the air is indicated directly in parts of carbon monoxide per 10,000 parts of air by a small millivolt meter, *G*, the pointer of which closes a relay circuit, *H*, when the concentration reaches any predetermined value at which the alarm is set. In its present form the instrument is intended more as an alarm than as a piece of control equipment, and as a result, it is so designed that when the concentration reaches the predetermined value and the bell starts ringing the circuit continues to be held closed by an electro-magnet until released by an attendant. Hence, in order to use the instrument in its present form as a control, it was necessary to develop a means for automatically releasing the contact at regular intervals of time so that, when the ventilating system, started by the control, had lowered the carbon monoxide content to within safe limits, the fans would be stopped. This was accomplished in the instrument used by the Laboratory by an electric circuit closed at one minute intervals, acting through a solenoid upon the pointer so as to release it. In an

instrument especially built for automatic control purposes, this feature can probably be better incorporated in the design of the instrument itself.

The arrangement of apparatus is shown in Fig. 4. The study was made in the two rooms formerly used by the Research Laboratory for investigating solar radiation through windows. A small kerosene engine was located in one of the rooms. The exhaust was so arranged that the gases could be directed to the outside or into the room, or divided between the two in any proportion by valve adjustment. For convenience of the attendant in making the necessary observations, the alarm was placed in the adjacent room, with the intake extended through the wall and into the room containing the engine. Means were provided for quickly taking a 250-cc sample of the air for analysis in order to check the accuracy of the indicator on the alarm and the point at which

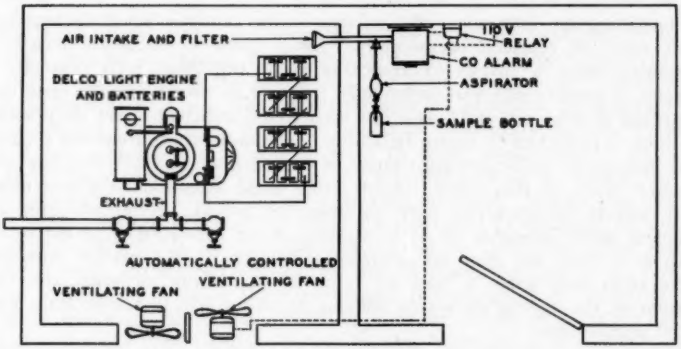

FIG. 4. DIAGRAM OF SET-UP USED IN STUDYING CARBON MONOXIDE ALARM

the alarm or control acted. The room to be ventilated was provided with two openings in the window, in one of which a small desk fan was placed to exhaust air. This fan was usually kept in operation and produced insufficient ventilation with a given rate of exhaust supply to the room to keep the carbon monoxide content of the air within safe limits, and as a result the carbon monoxide content would rise considerably above two parts, unless additional ventilation was provided. The second and larger fan was placed in the second opening in the window and supplied air to the test room at a rate sufficient to bring the carbon monoxide concentration down to one part or lower in about three minutes. This fan was operated by the control acting through a relay, so that as soon as the carbon monoxide reached the limiting value and closed the alarm circuit the fan was started and quickly brought the carbon monoxide concentration in the test room below the limiting value.

The instrument is designed to close the alarm or control circuit at approximately two parts of carbon monoxide in 10,000 parts of air. It does not have sufficient sensitivity to operate successfully and with sufficient reliability at a lower concentration.

The alarm and control was used in this set-up over a period of 30 days,

during which time over 50 samples of the air entering the instrument were collected and analyzed. The procedure in most of the tests was as follows: After the engine and automatically controlled ventilating system had been in operation for two or three hours and everything was operating smoothly, a sample of air was taken when the indicator on the alarm showed about 1½ parts of carbon monoxide in 10,000 parts of air. Another sample was taken when the indicator showed a concentration of 2 parts in 10,000, and a third when the concentration was a little higher. These samples were analyzed by the pyrotannic acid method.[8] Since this method of analysis is only sensitive to about ½ of a part of carbon monoxide in 10,000 parts of air, the analysis would only indicate the instrument to be accurate to within these limits, which was considered sufficient for the purpose. For all the samples analyzed, after the test procedure was well established, the indicator on the alarm was shown to be accurate to at least ½ part of carbon monoxide in 10,000 parts of air.

In operating the instrument the activity of the catalytic agent decreases so that it must be replaced occasionally. Since there is no ready means of determining when a charge of Hopcalite needs replacing, it is desirable to change it at regular intervals sufficiently frequent to insure reliable operation. Recharging at intervals of 14 days has been found sufficient under all practical conditions of operation to insure reliability. The only other attentions required by the instrument are occasional change of the air filter to avoid clogging with dust, and oiling of the motor. Aside from these attentions, the instrument should operate satisfactorily over an indefinite period. After charging the instrument with Hopcalite or after starting it up following a period of inoperation, it does not give an accurate indication of the carbon monoxide concentration until after a lapse of from 20 to 30 min, or until after the oxidation of the carbon monoxide and the resulting rise in temperature have reached equilibrium.

DISCUSSION

DR. E. V. HILL: I would like to know how close the alarm controls the fan, within how many parts per 10,000.

F. C. HOUGHTEN: It was desired to develop an instrument that would act when the carbon monoxide concentration approached one part in 10,000, for the reason that it is generally recognized that harmful physiological effects may begin slightly above this concentration.

The best that the instrument makers have been able to accomplish up to the present time is to develop an instrument which will operate at about two parts in 10,000, and will turn the fan on and off within limits of about one-half of one part in 10,000. Hence, if the alarm turns the fan on at two parts, it can be relied upon to turn it off at about one and one-half parts in 10,000.

H. S. WALKER[9]: Are those instruments in production? For what can they be purchased?

MR. HOUGHTEN: A few of these instruments are being made and sold; however, I do not know just what they cost. It was estimated that the instrument could be built to sell at $75.00, but this price has been creeping up as more refinements are added.

MR. WALKER: Who makes them?

MR. HOUGHTEN: The Mine Safety Appliances Co. of Pittsburgh, the same company which manufactures the U. S. Bureau of Mines Carbon Monoxide Recorder.

[8] Loc. Cit. See Note 5.
[9] Detroit Edison Co., Detroit, Mich.

No. 1005

THE NATURE OF IONS IN AIR AND THEIR POSSIBLE PHYSIOLOGICAL EFFECTS

By LEONARD B. LOEB* (*NON-MEMBER*), BERKELEY, CALIF.

THE recent focus of interest of engineers on the problems of air conditioning has raised the question as to the relative importance of various constituents of the atmosphere and their influence on the well-being of those breathing the air. Because of the general ignorance concerning the ionic constituents of the air, it is not unnatural that many persons should have ascribed somewhat miraculous powers to these constituents.[1] As a result, efforts have been made to invoke the gaseous ions in an attempt to explain certain of the less clearly defined influences observed. Needless to say, studies of such questions involve very carefully controlled experiments in which it is particularly difficult to eliminate and control the variables. The results are therefore to be regarded with a great deal of caution in view of the facts which are to follow. Actually the experimental investigations of recent years have fairly well clarified the understanding as to the nature of electrical carriers in gases so that definite conclusions as to their influence in air conditioning can be drawn. It is the purpose of this paper to present the facts as they are known today and to draw the obvious conclusions.

Carriers of electricity in gases, notably in mixtures such as atmospheric air, can be divided into three groups. These are the large or Langevin ions, normal ions, and free electrons. Early investigations of the large ions, notably the studies of Pollock[2] and Blackwood[3] have shown that these ions consist of material particles, dust or liquid droplets which in one way or another have acquired a charge. In this article, in order not to confuse the issue with too detailed a discussion, the term large or Langevin ions has been an inclusive one covering all carriers of electricity in gases having a velocity in unit field less than 0.1 cm per second per volt cm, and extending to the very large visible dust particles, which in virtue of being charged, are also in a sense to be classed as large ions.

* Dept. of Physics, University of California.

[1] Koller, L. R., *Jour. Franklin Inst.*, *214*, 543, 1932.
Yaglou, C. P., Benjamin, L. C., and Choate, S. P., Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 191.
Yaglou, C. P., and Benjamin, L. C., Diurnal and Seasonal Variations in the Small-Ion Content of Outdoor and Indoor Air, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 271.
Tchijievsky and Voynard, Compte Rendue, *199*, 496, 1934.

[2] Pollock, S., *Phil. Mag. 29*, 514, 636, 1915.

[3] Blackwood, O., *Phys. Rev. 16*, 85, 1920, and *20*, 499, 1922.

Presented at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by G. R. Wait.

Actually a distinction has been made between various groups of these ions by the men engaged in the investigation of atmospheric electrification, who subdivide this group into intermediate ions of velocities in unit field of 0.07 cm per second, Langevin ions of velocity 0.0005 cm per second, and very large ions. While there are detailed differences in the nature of the ions which are of importance to the students of atmospheric electricity, from the point of view of this article there is no essential difference, in that such ions, as will be seen below, can be considered in all cases as consisting of some sort of liquid or solid particle or nucleus which in some fashion has acquired a charge. A more detailed discussion of the nature of these ions and some of their properties will be presented by G. R. Wait in a discussion of this paper. Such particles, if solid, are usually covered with a condensed film of water vapor [4] whose presence and thickness depends on the degree of saturation of the air, and the nature of the particle (*i.e.* whether or not it is hygroscopic). As determined from their mobility (*i.e.* velocity in unit field) they vary in size from 10^{-7} cm in diameter on up (*i.e.* from aggregates of some thousands of molecules to visible particles). The number present varies with the condition of the atmosphere (*i.e.* on the amount of material in suspension) and on the ionic content of the air or else on the origin of the particles. If the particles are formed by the processes of combustion, by the spraying of liquids or by the agitation of material particles such as in dust storms in dry air, the percentage of charged carriers will be considerable. In the process of spraying liquids in general, it is believed, and in the case of water in particular it has been observed,[5] that the smallest particles are predominatingly negatively charged. In a recent study by Chapman [6] on drops of liquids which are generated by spraying and can be observed in a Millikan condenser, the signs of charges are independent of drop size and are the greater the larger the drops and the higher the dielectric constant of the liquid. Pure paraffine oil gives a few unit charges per drop for the smallest drops (about 3×10^{-5} cm diameter) and nitrobenzol yields some hundreds of charges per drop for the largest drops (some 4×10^{-4} cm diameter). In all but paraffine (where a few uncharged drops were found) practically all drops observed from 10^{-5} to 5×10^{-4} cm in diameter were charged and no one sign predominated. Whether the negative predominance on very minute drops (10^{-7} cm diameter) observed by Lenard and his school for water is universal for all liquids is yet to be determined. In the case of the very minute water drops only one in approximately 10^4 is negatively charged, which contrasts markedly with the case for the larger drops. The dusts and condensed vapors from flames and chemical processes have unquestionably a considerable number of charged particles of both signs. It is probable that in these cases the values of the charges per particle are small in number since they result from the picking up of ions (which are largely singly charged) or electrons from the reacting gases. Large particles are again more likely to have the more charges as they have a greater capacity and a larger collecting surface. In sand and dust storms the particles are charged through friction. The charges acquired are usually very high per particle and if the air is dry and turbulent the numbers are also very large.

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[4] Loc Cit. See Note 2.
[5] Lenard, P., Ann. der Phys. *47*, 463, 1915.
[6] Chapman, S., Physics *5*, 150, 1934.

In *ordinary* air unless the humidity is low and turbulence is very great, the large ions are few in number. Doubtless some of the particles occasionally pick up charges from the very few ions normally present and become charged. The majority are uncharged. Masses of condensed water vapor such as mists or fogs far from industrial contamination are largely uncharged. Sea fogs, while containing notable amounts of dissolved salts from spray action, show little indication of charge since the natural ionization of the air at sea is distinctly smaller than over the land. Mists formed inland which contain as nuclei particles of dust and products from industrial activity may be somewhat more highly charged, but their charges are minute compared to those in dust storms.

Whether the presence of charges on solid or liquid particles in the atmosphere exerts any physiological action is not known but seems highly questionable. These particles are inhaled with the air and either reach the mucosa of the bronchial tract or enter the alveoli. All these regions of the body are covered with a conducting liquid film which either permits the neutralization of ions of opposite sign or allows the charges, if of one sign, to leak off to ground through the body; for the body, regarded from the point of view of small quantities of static charge, such as under consideration, is a fairly good conductor. If ions of one sign only reached the lining of the lungs, they would of course pass a current through the tissue walls on their course through the body to ground. Such galvanic currents in cells, if of sufficient magnitude, could produce physiological effects through electrolysis. These static currents are, however, of such magnitude that even were the ions all of one sign (a condition rarely realized), the effect could never be noticed, since the amount of solid or liquid matter required to introduce charges of any magnitude must in themselves produce pronounced and masking physiological effects. This conclusion becomes obvious if one realizes that if 1 cm^3 of dust particles of 10^{-4} cm diameter were inhaled in 100 seconds and if 1 per cent of these were all of the same sign and charged to 1000 electrons each (a not unlikely charge value in a dust storm though the predominance of sign does not exist) the current passing through the body would be about 3×10^{-8} amperes and the current density (as this is spread over an enormous area of mucosa) would be so small as to produce no noticeable physiological action even if it all passed through the epithelial cells. On the other hand, what the physiological effects would be following the inhalation of uncharged dust at the rate of 1 cm^3 in a minute and forty seconds can easily be imagined. Similar reasoning can be applied to sprays of such a harmless substance as water where, however, it is found that only one in 10^4 of the minute droplets is predominatingly negatively charged and that to only unit charge. One is thus led to conclude that the physiological effects produced in the inhalation of large ions must be immeasurably small compared to the effects produced by the accompanying solid or liquid carriers on the respiratory tissues. As regards the ions to be obtained in inhaling fumes from chemical reactions such as combustion, it is clear that the ions, which are far less dense than in the cases previously cited, would be accompanied by such amounts of toxic substances that further discussion is unnecessary.

Normal ions are produced in air as a result of the actions of various radiations incident on air molecules or solid particles in contact with the air. These

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radiations consist of cosmic rays and their secondaries (always present but very few in number), gamma rays from radioactive bodies, X-rays, and ultraviolet light. Gamma radiation is always present in some amount from the radioactive constituents present in the earth and air, but is relatively weak over the ocean. X-rays normally are not present in any quantity except in laboratories. Ultraviolet light is present in sunlight but the wave lengths are too long to ionize the air appreciably at the surface of the earth. Terrestrial sources such as quartz mercury arcs or open arc lamps give some small amount of radiation capable of ionizing the gas molecules; these arcs also are capable of causing fairly intense electron emission from metal surfaces. Ultraviolet radiation can thus be neglected as a source of atmospheric ionization under usual circumstances. In addition, the alpha and beta rays of radioactive substances can ionize the air very effectively. They come from emanations of radium and thorium present in the ground in minute amounts which escape as gases from the soil. These account for some 6 to 14 of the ions formed per cm^3 per second in ordinary air over the land, while cosmic radiations account for some 1 or 2 and gamma rays for 2 to 4 of the ions formed during the same time. The presence of electrical discharges, lightning, corona from high tension lines, sparks, etc., produces intense ionization near the ionized paths. Also the gases from flames and near certain chemically reacting gases may have a relatively high content of gas ions. Thus aside from the air in the neighborhood of artificial specifically ionizing agents, the normal atmosphere is little ionized. Even in industrial centers, the rate of normal ion production in the air rarely exceeds 20 ions per cm^3 per second, and is more nearly 10. Owing to diffusion and recombination, these ions disappear at a rate which is not negligible so that the ionic content of the air is normally never very much greater than some 2000 ions per cm^3, and usually about 1000. Actually, owing to the presence of dust particles, water vapor nuclei and other suspended material in the atmosphere, the small ions which are produced are very quickly picked up, in some cases through their charge stabilizing the nuclei or droplets of water vapor and insuring their continued existence as large ions so that actually the number of small ions existing in air at any time is approximately a fifth to a tenth of the number mentioned, *i.e.*, 200 small ions per cc is a proper order of magnitude. It has been shown by Wait [7] that certain changes in the small ion content in rooms occupied by people as shown by Yaglou [8] and his collaborators can be explained as being due to the fact that in occupied rooms or rooms having large numbers of nuclei present, the small ions are quickly picked up to form the larger classes of ions and hence the number of small ions in a room will vary with the degree of occupation and contamination. Again, it is possible that owing to the very much reduced recombination rate of the larger ions, *i.e.*, those of mobility 0.0005, the number of charged particles per cc comprising both small and large ions may be somewhat greater than the 2000 ions per cm^3 mentioned above. This will be especially so where there is much industrial contamination, and possibly as many as 10,000 ions per cm^3 will occur on occasion. Relative to this, the total number of nuclei of all sizes may reach as many as 100,000 per cm^3. In fact, even with intense X-ray ionization in the laboratory, it is hardly possible to exceed concentrations of 10^6 or 10^7 ions per cm^3 at any one time. While arcs and elec-

[7] G. R. Wait, *Journal of Industrial Hygiene, 16,* 147, 1934.
[8] Loc. Cit. See Note 1.

trical discharges have far higher ion concentrations in their paths, this ionization cannot be widely diffused through the air in high concentrations. X-ray ionization thus probably furnishes the greatest volume ionization, without great contamination of foreign particles, obtainable. If the ions are produced in the air, the positive and negative ions appear in equal amounts, while ionization produced from solid surfaces by X-rays will have primarily negative ions.

The nature of the normal ions in air is today fairly well known. Ionizing agents detach from the oxygen or nitrogen molecule as a rule a single electron.[9] On the average this electron has a considerable excess of kinetic energy given it by the ionizing agent and thus escapes rather far from its original atom in terms of the scale of atomic dimensions. Except at pressures of hundreds of atmospheres, the electron very rarely, if ever, regains its parent molecule. It usually moves about quite rapidly in the gas, gradually losing its excess energy until a fortuitous encounter with an oxygen or some complex molecule causes it to attach to that molecule to form a negative molecular ion.[10] Not much is known about this process except that in O_2 it takes some 10^4 impacts on the average to attach, although this figure varies with its velocity,[11] increasing as the velocity increases. The heat of formation of such an ion is of the order of 0.4 volt of energy,[12] (*i.e.* about one-tenth the heat of dissociation of H_2) and presumably gives a radiation well in the red end of the spectrum. The positive molecular remainder after the electron has left it may dissociate in some gases. Usually it does not. If multiple charged by the occasional loss of two or more electrons in the ionizing process, it will in a short time steal electrons from a neutral molecule making as many singly charged ions as it had positive charges.[13] While multiple charged ions are sometimes formed, there is no evidence of their existence after the thousandths of a second or more during which ions are studied. This is not strange as ions make some 10^9 impacts per second in a gas and even if only one impact in 10^6 resulted in some event, the event would occur within 10^{-3} seconds. More often if much impurity is present, the O_2^+ ion may steal an electron from some molecule of slightly lower ionizing energy in an impact giving a new charged nucleus and O_2. This has been observed to occur in gases by Kallmann and Rosen,[14] and for ions by Luhr.[15]

As is well known, ions in solution have very specific electrochemical affinities for some molecules, making the so-called complex ions. Such ions as the NH_4^+, the $Cu(NH_3)_4^{++}$ and the $CU(CN)_2^-$ ions are common examples. Studies of the behavior of gaseous ions in mixed gases and of the aging of ions have demonstrated beyond doubt that such combinations also occur in gases. Thus it does not surprise one to find the positive ion attaching readily to NH_3 or RNH_2 compounds to form complex ions in gases. This also occurs

[9] Loeb, Nature of a Gas, John Wiley & Sons, New York, 1931, p. 104.
[10] Loeb, Kinetic Theory of Gases, 2nd Edition, McGraw-Hill, New York 1934, Sec. 109, p. 613 ff.
[11] Bradbury, N. E., *Phys. Rev. 44*, 883, 1933.
[12] Loeb, L. B., *Phys. Review, 48*, 684, 1935.
[13] Loc. Cit. See Note 9.
[14] Kallmann and Rosen, *Z. Physik. 58*, 52, 1929; *61*, 61, 1930; *64*, 806, 1930.
[15] Luhr, O., *Phys. Rev. 44*, 459, 1933.

for these ions with ethyl ether and the cyanides.[16] In contrast, the negative ions pick up molecules of Cl_2, alcohols, ROH and both ions pick up HCl and SO_2. It was long believed, due to the fact that observed ionic mobilities in gases were about one-fifth what one would expect did their charge not affect their motion by attracting neutral molecules, that the ions were *clusters* of some 6 to 30 neutral gas molecules. The discovery of the formation of the specific ions described above and much other evidence has made the large cluster ion seem unlikely,[17] and it is probable that the ions are a positively or negatively charged molecule which in the right environment has been able to pick up some especially powerfully attracted impurity to form a complex. From the stability of these complex ions and other knowledge, it is unlikely that many more than 3 or 4 of the active molecules enter into the ion complex. This is in essence not a cluster. The further decrease of the mobility formerly ascribed to a large cluster is completely accounted for by the impaired motion of the ion through the gas, in view of its dielectric attraction for the gas molecules, which is, however, too weak and non-specific to cause clustering. The Debye-Hückel theory of strong electrolytes in solution is based on an analogous viewpoint. This fact is borne out admirably in recent experiments of Tyndall, Powell and Brata.[18] Thus the low mobility can be explained by the inverse fifth power law of dielectric attraction, while the behavior in mixed gases, aging, and the differences in mobility of positive and negative ions in different media are conditioned by complex ion formation.

One may well ask what one can expect to find in the complex ion groups in air. This depends upon age, on impurities present and on the intensity of ionization in air. Luhr [19] has observed the masses of ions in very pure N_2 and O_2 very shortly after formation. In N_2, the ion will, if given time, add another N_2 molecule or an N atom, giving N_4^+ and N_3^+. In O_2 one has only O_2^+ and a few O^+ ions. In air these ions occurred as well as those of the oxides of nitrogen. The transfer of charge noted above was found to occur from O_2^+ to NO, NO_2 and N_2O and especially to water vapor present. This referred to ions some 10^{-6} seconds after ionization. If, however, NH_3 or Cl_2 or SO_2 are present, even in traces, these simpler complexes give way to the much more stable complexes of NH_3, Cl_2 and SO_2. In fact it has been shown by Bradbury [20] that the degree of purity essential for mobility studies on uniform and reproducible ions requires techniques which transcend any chemical techniques known. For example, O_2 prepared by methods used in the most accurate atomic weight determinations were useless for ion work and other techniques had to be resorted to. Since 10^9 impacts occur in one second if only one part in 10^6 of a substance is present which can replace another molecule of a complex ion in one out of 10 impacts, a mobility method giving results on ions 10^{-2} seconds old (most methods used take this time) will show the more stable

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[16] Loeb, L. B., *Phys. Rev. 32*, 81, 1928.
Loeb & Dyk, *Proc. Nat. Acad. Sci. 15*, 146, 1929.
Loeb, L. B., *Phys. Rev. 35*, 184, 1930.
[17] Loeb, Kinetic Theory, Loc. Cit. See Note 7. Sections 104, p. 55, and 106, p. 564.
Loeb, L. B., *Phys. Rev. 38*, 549, 1931.
Loeb & Bradbury, *Phys. Rev. 38*, 1716, 1931.
Loeb, L. B., *Zeits. f. Phys. 75*, 555, 1932.
[18] Tyndall & Powell, *Proc. Roy. Soc. A136*, 145, 1932.
Powell and Brata, *Proc. Roy. Soc. A138*, 117, 1933.
[19] Loc. Cit. See Note 15.
[20] Bradbury, N. E., *Phys. Rev. 40*, 508 and 524, 1931.

impurity complex as predominating. Hence in a gas like air, the ions will be progressively changing the nature of their complex such that in sufficient time all negative and positive ions will be complexes having the largest and more bulky but stable impurities as companions. Such progressive aging has been observed,[21] and the tendency of the larger and more bulky organic molecules to form the more stable complexes is an established fact.[22]

Thus in air all one can say is that the ions will consist of groups of varying ages and complexity whose most frequent nature will depend on the stability of the complexes possible and the impurities present. As air is a complex mixture which under the intense ionizing radiations has the chance of forming compounds having high affinities, it is not surprising to find that such products influence the mobility markedly. This has been observed by Erikson [23] and by Varney.

When one considers the physiological effect of normal ions, one must distinguish two possibilities: the effect of the charge on the mucosa and the tissues, and the effect of the foreign molecules brought into the body by the complex. As has been indicated in the case of the large ions, the effect of the charge cannot be significant. This is especially true since the greatest charge density which can be effectively produced and maintained conveniently in air is 10^7 ions per cm^3, which would give currents of 3×10^{-9} amperes if two liters of air were inhaled per second and all ions *were absorbed and were of one sign,* which does not occur. Ordinarily, of course, one must realize that normal air has of the order of 10,000 or distinctly less carriers of all sizes including visible dust particles per cm^3 in a total of 2.7×10^{19} molecules of gases composing the air. Hence even impurities which cannot be detected chemically, *i.e.* present to one part in 10^7, will have 10^{12} molecules present in contrast to 10^4 ions. This makes the effect of the variation of ionic content in normal air as a health factor seem ridiculous in the extreme.

In discussing the companions of the ions, one faces another question. Could one control the age of the ions breathed and the superchemical purity of the air, one might get a selective group of molecules into the respiratory tract. The amounts would, however, be exceedingly small compared to the quantities of the same products which could be inhaled by introducing one part per million of the chosen molecules into the air that is inhaled. Hence the value of gas ions and in fact their effect, would be ridiculously minute. Experiments in this direction are hardly needed as simple calculation reduces the question to one of absurdity.

On the other hand, an attempt to study this question experimentally on human subjects (as has been done [24]) ceases to be one of absurdity. It is exceedingly dangerous to health. The danger lies in the following fact. While ions are relatively unstable entities (*i.e.* they diffuse, give up their charge to solid particles and rapidly recombine), all ionizing processes in air give rise to chemical reaction products of marked physiological activity which are millions of times more stable. Thus in one second, one-half of the million ions initially created in a cm^3 of air have recombined. In the same cm^3, while 10^6 ions were formed, it is not rash to estimate that some 10^7 or 10^8

[21] Bradbury, N. E., *Phys. Rev. 37,* 1311, 1931.
[22] Loc. Cit. See Note 16.
[23] Erikson, H. A., *Phys. Rev. 34,* 635, 1929.
Varney, R. N., *Phys. Rev. 42,* 547, 1932.
[24] Loc. Cit. See Note 1.

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molecules of various sorts such as ozone, N_2O_4, NO_2, NO, etc. have also been formed. These are stable, even ozone lasting many minutes. It is seen then that in an attempt to create a few million ions per cm^3 and to maintain that concentration for inhalation experiments, the air in a very few minutes is charged with concentrations of violently reactive gases transcending ion concentrations by many thousands of times. That such must be the case is seen when one realizes that ionization is a very inefficient process; for about half the energy put into ionization in air (16 volts as against 33) goes into ionization, while the rest is spent in activating the air and energy of activation for most chemical processes is a small fraction of the ionizing energy. Finally, even the more unstable chemical products are thousands to millions of times more stable than the ions.

Concerning the danger of these products to human health when taken into the lungs, little need be said. Ozone was at one time heralded as a vitalizing and health-giving product, a *cure-all* for disease and a wonderful disinfectant. The medical profession has long ago abandoned ozone as a particularly irritating and dangerous compound.[25] The small amount of ozone created by a quartz mercury arc in a confined room during an hour or two a day for a few weeks has been a contributory cause of serious chronic respiratory afflictions in several cases to the knowledge of the writer. Ozone is a germicide and thus may be as destructive of human tissues as it is to bacteria. As regards the oxides of nitrogen, the physiological effects, especially of those leading to nitric acid formation, are self-evident. Even moderate exposure to such products as produced by the high tension brush discharged from improperly protected power lines in a confined space may contribute to the more serious pulmonary infections such as pneumonia. In fact, it should be emphasized that the present occasional practice of *purifying* the air of theaters and public halls[26] by means of O_3 and its accompanying nitrogen oxides or by Cl_2 gas should be terminated until very much more study has been made of the effect of small amounts of these substances in air on health.

As regards free electronic carriers, little need be said in reference to their possible physiological action in air. The free electrons, no matter how generated, in some 10^{-4} or less seconds will attach themselves to neutral molecules to form negative ions. They do not exist in air for a sufficiently long time as electrons to be considered as a feature at all. Thus the effects which they might cause must all be included under the heading of ions which have already been discussed.

It is obvious from the foregoing that the possible physiological effects of gaseous ions as they influence the problems of air conditioning are absolutely negligible. Not only that, but any attempts to investigate such influences experimentally on human beings must not only be regarded most critically but must also be regarded as dangerous. For in order to introduce appreciable charges into the respiratory tract to produce galvanic currents of any magnitude, so much foreign material, solids (dust, MgO, etc.), liquids (sprayed water dust), or secondary reaction products such as O_3, N_2O_4 or NO_2 must be taken

[25] McDonnell, H. B., *Jour. Assoc. Official Agric. Chem. 13,* 19, 1930.
Marek, Karl, *Zeits. Hyg. 81,* 221, 1915.
Marek, Karl, *Chem. Zeits.* II, p. 188, 1916.
Toul, Fr., *Casops Ceskaslov Lekanictva, 60,* 264, 1930.
[26] See *Gesundheits-Ing.* vol. 53, p. 499, 1930.

into the lungs as almost certain to produce pneumonia, severe pneumonoconiosis or other chronic pulmonary ailment, and to favor respiratory infection in general. It may be added that many of the chronic pulmonary ailments produced by inhalation of foreign materials have incubation periods of such length that the effects of such inhalation may not at once become obvious. For example, the incubation period of silicosis [27] has recently been established as some 12 years. In view of the great predominance of foreign matter in air compared to the indefinitely minute ion concentrations, the air conditioning experts would do far better to study and understand the effects of small amounts of dust, ozone, N_2O_4, SO_2, NH_3, CO_2, CO and water vapor in the air on the comfort and well-being of the inhalants, than to waste any time in considering gas ions.

DISCUSSION

C. P. YAGLOU (WRITTEN): Professor Loeb sounded a most timely warning against the possible dangers involved in certain ionizing processes which give rise to toxic by-products in harmful concentrations. In so far as I know, his physics is perfectly sound and the only differences I have in this respect are expected deviations of practical realities from fundamental physics, having to do with such factors as ion density, unipolarity, and control.

The more important questions in my mind concern: (*a*) the possible physiological action of ionized air, and (*b*) the possibility of danger from toxic by-products in concentrations in which they actually occur under proper conditions of artificial ionization.

I agree entirely with Professor Loeb in regard to the improbability of expecting demonstrable physiologic effects from the insignificant static charge or galvanic currents which the tissues may receive from the inhaled ions. An important question which the author overlooked is the possibility of radiation effects. When positive ions strike the epithelial cells, electrons fall from outer orbits to inner orbits and the ions are in a state of radiation, the spectrum of which may be looked for in the ultraviolet. To this radiation we are inclined to ascribe the irritation and dryness observed by us among a group of sensitive subjects after exposures of 1 to 2 hours to small positive ions in concentrations of 1,000,000 per cc or more. On prolonged daily exposures to such ion densities both my colleague and myself developed a mild bleeding in the nose and throat which cleared up only after stopping work with positive ions. In so far as we know, this is the only harm that may result with suitable apparatus when the ionization is positive.

The effects do not occur with negative ions, and we two cannot see how negative ions can produce any physiological effect, except perhaps indirectly by precluding radiations from the positive ions which are normally present in air.

A question may arise whether the irritation and bleeding cannot be attributed to ozone and oxides of nitrogen. Investigations of this problem were made during the development of apparatus. Standard tests by the sense of smell and by the potassium iodide method failed to detect ozone, even in a concentrated stream of air 1/200 of the usual amount of air passing over the ionizer. Since the practical limit of accuracy of the potassium iodide method is about 0.1 parts per million, simple computation will show that whatever amount of ozone is produced by our process must be in concentrations less than 1 part in two billion parts of air by volume. On the basis of the olfactory threshold (0.0001 to 0.002 parts per million) it must be lower than 1 in 100,000 billion parts. Outdoor air, especially open country air, contains much more ozone than this as will be indicated later.

We did not make special tests for oxides of nitrogen for reasons that are too

[27] Pneumonoconiosis-Silicosis, *Journal of the American Medical Association, 101,* 580, 1933. Gardner, *Journal of the American Medical Association, 101,* 594, 1933.

obvious. Nitrous fumes are not produced in detectable quantities in the absence of sparks from ionizing equipment. Even in commercial ozonators which produce ozone by sparks, Jones and Yant found that the ratio of the oxides to ozone produced is about 1 to 150.

I cannot agree at all with Professor Loeb that the chemical reaction products of ionizing processes described by him as O_3, N_2O_4, NO_2, NO, and N_2O are dangerous to health when present in air in concentrations of 10^8 molecules per cubic centimeter, or 1 part in 271 billion parts of air.

The physiologic action and degree of toxicity of ozone and oxides of nitrogen are fairly well known. To begin with, NO may be dismissed entirely from Professor Loeb's picture because it does not exist in air. It combines immediately to the dioxide in the presence of air or oxygen. Also N_2O (laughing gas) may be dismissed from the picture because it is not an irritant. Like CO_2 it is a simple asphyxiant and deserves less consideration than CO_2 owing to its comparative rarity in air. It is used extensively in anesthesia in concentrations as high as 90 per cent, the remaining 10 per cent being pure O_2.

The olfactory threshold of O_3 is placed at from 1 part in 10 billion to 1 in 500 million, or 27 to 540 times the concentration which Professor Loeb considers exceedingly dangerous. According to the literature (Hill and Flack, and others) there is no harm in breathing such weak concentrations of O_3. As a matter of fact pure country air under normal meteorologic conditions may contain as much as 1 part in 700,000 parts of air by volume (Rosenau) or about 400,000 times the concentration considered extremely dangerous by Professor Loeb. Polar air contains even more. City air on the other hand, contains less than country or polar air (except during local thunderstorms) and the amount is difficult to estimate, although under favorable conditions it can easily be detected by the sense of smell. The irritation threshold is 0.5 to 1 part per million or 140,000 times Professor Loeb's concentration. At between 0.5 and 1 per million, the oxygen consumption and CO_2 production begin to diminish (Bohr and Maar) and after some time cough and fatigue may result (Lewen). The pulse and body temperature after a slight initial rise return to normal (Felipow).

Nitrous fumes in the form of NO_2 and N_2O_4 are not considered as powerful irritants as O_3, although they are the most insidious of all irritant gases. The maximum concentration of nitrous fumes allowable for prolonged exposures is 39 parts per million (Lehman and Hasegawa) or about 10,000,000 times greater than Professor Loeb's dangerous concentration.

It is hardly necessary to go farther into the literature to prove the fallacy of expecting any harmful effects from ozone and oxides of nitrogen under the concentration estimated by Professor Loeb himself.

In the light of these authoritative facts, I should like to ask Professor Loeb to reconsider his view and, if justified, to modify the dogmatic stand which he has so arbitrarily taken. He has done a great deal of work in related fields of pure physics and we need his advice and constructive criticism.

Before closing, I should also make my own stand clear. From the viewpoint of general ventilation, I agree with Professor Loeb that ionization is probably not an important factor. The conclusion we have reached is that "nothing was found in our experiments to justify the use of artificial ionization in general ventilation."[28] Probably Professor Loeb did not see this paper. The reference he gives in his bibliography is probably an error since it deals with diurnal and seasonal changes in atmospheric ionization and has nothing to do with physiologic or any other effects.

Dessauer's and our experience with abnormal persons and patients of the clinic does not justify dismissing the matter so easily as Professor Loeb wishes us to do. So long as harm can be precluded, I am firm in the attitude of having the whole question left open until more data are obtained under properly controlled conditions.

[28] *J. Ind. Hyg.*, Sept., 1933, *15*, 5.

L. B. LOEB (WRITTEN): In a discussion of this paper Professor Yaglou drew certain inferences which are clearly not stated in the paper. In view of the misrepresentation of what I stated and believe, as a result of these incorrect inferences, a few words on my part are in order.

1. In the first place, there was a typographical error on page 108 in the twelfth line from the end of the original article in which the 2 of N_2O was obviously transposed from NO_2. It is hardly likely that even a physicist would have called laughing gas, N_2O, a toxic product.

2. Professor Yaglou quotes me as saying that in concentrations of 10^8 molecules/cm^3 the reactions products under discussion are injurious to health. Any one reading this paper will note that no such statement was made, nor was any claim made as to the toxic concentrations at any point. Quoting from page 107: "The danger lies in the following fact. While ions are relatively unstable entities (i.e., they diffuse, give up their charge to solid particles, and rapidly recombine) all ionizing processes give rise to chemical reaction products of marked physiological activities which are millions of times more stable. Thus, in one second, one half of the million ions initially created in a cm^3 of air have recombined. In the same cm^3 in which 10^6 ions were formed, it is not rash to estimate that some 10^7 or 10^8 molecules of various sorts such as ozone, N_2O_4, NO_2, NO, etc., have also been formed. These are stable, even ozone lasting many minutes. It is seen then that in an attempt to create a few million ions per cm^3 and to maintain that concentration for inhalation experiments, the air in a very few minutes is charged with concentrations of violently reactive gas, transcending ion concentrations by many thousands of times. That such must be the case . . ." etc. This statement does not give 10^8 molecules of impurities per cm^3 as the concentrations used in treatment, in fact a sentence or two further it states, "transcending ion concentrations by many thousands of times" (ion concentrations having previously been given as 10^6 or 10^7 ions per cm^3). In fact at no point was it referred to as dangerous concentrations but it was merely stated that "in order to introduce appreciable charges into the respiratory tract to produce galvanic currents of any magnitude so much foreign material . . . or secondary products such as O_3, N_2O_4, or NO_2 must be taken into the lungs as almost certainly to produce . . ." The statement must be taken as it stands and is intended without the inferences Professor Yaglou draws. In fact, as regards the significance of the 10^8 molecules of impurities per cm^3 they were determined as the number that might be formed per cm^3 when 10^6 ions per cm^3 were formed. As these products are stable while ions are not, by the time the concentrations of millions of ions reach the lungs, the ratio of the concentrations will be many times the ratio of 1 to 100 given.

Each method of ion production and inhalation involves different conditions. Anyone would be rash to estimate the concentrations involved. They can and should be determined. It was and is my contention that the concentrations of foreign and probably physiologically active constituents in the atmosphere accompanying attempts to create ions are generally so great compared to the number of ions that we should rather look for, identify and study the effects of these, before we ascribe any effects to the gaseous ions.

3. The discussion of injurious concentrations of substances and the concentrations of these substances accompanying the ionization of the air used in experimentation by Professor Yaglou are interesting. It is my impression however that the arguments he sets forth concerning the concentrations used by him are neither satisfactory nor convincing. It need hardly be pointed out that such an individually variable and subjective test as the olfactory test for ozone is not quantitative. I am also uncertain as to the potassium iodide test with such minute amounts unless it is very carefully conducted. The problem involved in such small concentrations as are involved here lies in being sure with the slow diffusion that these few gas molecules have time to reach the absorbing medium and thus to be counted. Pro-

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fessor Yaglou states, "We did not make special tests for oxides of nitrogen for reasons that are too obvious. Nitrous fumes are not produced in detectable quantities in the absence of sparks from ionizing equipment. Even in commercial ozonaters which produce ozone by sparks, Jones and Yant found that the ratio of the ozone to oxides produced is about 1—150." If the interpretation of the second sentence is that oxides of nitrogen are not formed in ionizing discharge processes other than in sparks I cannot agree that this statement is in agreement with facts. Anyone who has worked to any extent with corona, brush or analogous discharges in an atmospheric pressure is well aware that this is an excellent source of these oxides. In fact it still remains to be proven that such oxides are not produced even in the illumination of air by light from a quartz mercury arc. We have evidence of quite a definite sort from Varney's work on mobility spectra in air that these products are formed.[29] There is again no reason why under conditions producing ionization in air by any means nitrogen oxides should not be formed in quantities comparable to ozone, since in addition the ozone is unstable.[30] If the statement made by Professor Yaglou is substantiated by tests it would appear that the methods of estimating small quantities of these substances need serious revision. In the absence of details as to how Professor Yaglou obtained his ions it is impossible to estimate the quantities of foreign materials produced, and it appears that he himself cannot say with certainty. This admission by him is exactly in line with the theme of my original article. Ions are worked with, apparent effects are attributed to ions (which in my belief are too few in number to produce effects) without any proper controls being made to ascertain that the effects if present[31] could not have been produced by other agents.

4. As a physicist I am quite interested in an attempt by Professor Yaglou to explain the effect of positive ions in causing "dryness and irritation." He explains these unpleasant effects as *possibly* being caused by an ultra-violet radiation due to electrons falling into the inner orbits on the neutralization of the positive ions. I had not considered this possibility in regard to the action of the positive ions in my paper as it never occurred to me that this very slight effect could be invoked. It is perhaps worth while that I present a few definite and undeniable physical data on this effect.

Assuming the improbable best case for Professor Yaglou's argument of 10^7 ions per cm^3 and an ionizing potential of 16 volts the radiant energy produced by the 2×10^{10} ions in two liters of air inhaled per second would be $2 \times 10^{10} \times 16 \times 1.6 \times 10^{-12} = 5.1 \times 10^{-1}$ ergs per second if it all went to radiant energy. This radiation if all in the ultra-violet of about 3000Å assuming no degradation of energy took place, (which is far from true), would be in the form of some 5 quanta per ion. Assuming that all the radiant energy fell on the surface of a two liter container (actually it would fall in the lung capillaries at a very much smaller surface density due to the large surface per unit volume exposed), the incident energy density in some 10^{11} quanta in the ultra-violet radiated per second would be $\frac{5.1 \times 10^{-1}}{750} = 6.8 \times 10^{-4}$ ergs per cm^2 per second. To the physiological and chemical action of this radiation density, Professor Yaglou as an hypothesis ascribes his symptoms. As a matter of fact, were all the above radiation liberated in the visible region with perhaps 10 quanta per ion, the light could not be perceived visually. Visual effects would require many thousands of times this energy. In the Smithsonian Tables, 8th revised edition, first reprint, March 1932, page 555, is given the density of ultra-violet solar radiation above 3130 Å on the clearest days for various hours of the day and months of the year. Let us take the noon radiation in June which is 82×10^{-5} cals per cm^2 per minute. This could, on the sensitive skin of a normal person, perhaps produce an erythema

[29] *Phys. Rev. 42*, 547, 1932.
[30] Loc. Cit. See Note 15.
[31] Hetherington, *Journal Clinical Investigation* XIV, 70, January 1935.

in some 10 or 15 min exposure. The energy in this radiation calculates to $\frac{82 \times 4.18 \times 10^7}{60 \times 10^5} = 5.7 \times 10^2$ or 570 ergs per cm² per second. Even on poorer days the amount between 10 a. m. and noon is not much less than ½ to ¼ of this. Thus the ion radiation to which Professor Yaglou attributes the possible physiological effect is $\frac{6.8 \times 10^{-4}}{5.7 \times 10^2} = 1.2 \times 10^{-6}$ or about one one-millionth as intense as the ultra-violet radiation in normal sunlight at noon or hundreds of thousandths as intense as the ordinary solar ultra-violet intensity to which we are normally subjected for 4 to 6 hours daily during the year. Actually wave mechanics has shown us that positive ions are very rarely neutralized by electron capture with radiation. Practically all ions are lost by recombination with opposite ions or with surfaces. These are *non radiative* processes. It is doubtful whether the ions produce 10^{-6} of the amount calculated above. In the case of MgO ions there is *no radiation*. Further comment is hardly needed. Scientific conclusions cannot be based on theories whose quantitative approximations to reality disagree in many orders of magnitude.

5. There are doubtless many unknown factors affecting health and comfort. Many of these may be inherent in the air we breathe. A carefully controlled series of investigations as to the nature of these unknown factors must be made. Possibly some of the factors may lie in secondary substances present in the air. The quantitative facts speak strongly against the ions, so strongly to one who sincerely analyses the factors involved that he should look elsewhere for the causes of the observed effects of this nature. It certainly appears unwise with the present status of our knowledge to experiment on human beings with these agencies unless exceptionally complete and carefully controlled. It was with the express purpose of calling attention to the glaring discrepancies between the quantitative facts in this field and the assumed explanations of the phenomena that the article was written. It seemed the more timely inasmuch as the results of certain experiments by legitimate investigators in this direction were giving the nostrum vendors an unusual opportunity not only to deceive the public but possibly to do unlimited harm in the premature application of various forms of treated or ionized air to ventilating engineering.

DR. C. P. SCHNEIDER[32] (WRITTEN): I have read the author's paper with interest and wish to discuss it from the standpoint of a doctor of medicine.

It is borrowing from the limbo of what might be called the folk-lore of medicine to state that the sensations of health and disease are affected by the weather.

So universally accepted a common statement must have some basis in fact—and, after years of lack of acceptance of such possibility by orthodox medicine, is now being recognized as true. Much more must be known, however, as to the mechanism.

Research in air conditioning has been an attempt to reproduce the desirable features of natural atmospheric conditions in order to make people more comfortable. In this respect the respective fields of the heating and ventilating engineer and the physician overlap.

To date, the regulation of better known features in air conditioning, such as temperature and humidity, leaves us short of the goal—and it is easy to understand why we are drawn to the possible utilization of ions—especially with the dawning appreciation of the electro-physical nature of all natural phenomena, including physiology. Such experiments to date, as we have knowledge of, do lend confirmation to this theory.

We are all aware of the definite changes that occur in an occupied room—characterized by a sensation of stuffiness—even though the temperature and humidity of that room are carefully controlled.

It would seem that if ionized air has physiologic importance, it must be due to

[32] Medical Dept., Detroit Edison Co.

the charge, minute as it might be. We do not know what magnitude of current nature employs in producing minute changes in cellular physiology. The charge on the surface membrane of a microscopic cell cannot be of very great magnitude and yet is of vital importance in the physiology of that cell. Such a charge, for instance, determines the secretion of hydrochloric acid by cells in the stomach.

It should not be necessary to "introduce appreciable charges into the respiratory tract to produce galvanic currents of any magnitude" in order to produce a physiologic effect. Subjectively desirable natural atmospheric conditions seem to contain 7 or 8×10^3 ions per cm^3. It is questionable that one would need the high concentration of 10^7 per cm^3 that the author speaks of. Large ions do seem to carry the disadvantage of large quantities of matter but it should be possible practically to eliminate the large ions in experiments.

Professor Loeb seems to feel that effects can be produced only by absorption of ions from the lung; the large area of upper respiratory mucous membrane as well as the possibility of absorption from the skin surface must not be lost sight of.

The author mentions the harmful effects of ozone and oxides of nitrogen. It has been found possible to produce ions without simultaneously generating harmful gases and we know that ions exist in nature without harmful concentrations of such gases.

With these potentially harmful features eliminated, it would seem that with careful control and with preliminary experiments on animals, it should be possible to produce satisfactory experiments on human subjects without harmful effects.

From a physician's standpoint alone, I feel very strongly that such research, offering as it does such unlimited possibilities, cannot and should not be condemned on theory alone.

Finally one must remember that the author has reached conclusions from a priori reasoning and not from experiment. There is in existence certain experimental evidence which tends to refute his arguments. There are all too many instances in history where scientific advancement has been retarded by just such reasoning.

J. J. AEBERLY (WRITTEN): The author's presentation sets forth in a logical and easily understood manner some of the physical problems involved in the question of ionization of air and to this extent it may be stated that a clearer understanding of the physics involved has been presented to the membership as a whole. The author has particularly stressed the minuteness of the quantities of the various kinds of ionized matter which may be found in air, and from this has assumed that because these quantities are minute there will result little or no physiological reaction.

At this time it may be well to review the purpose of these studies in ionic content of air and their physiological reaction. These were undertaken by the Society as part of a comprehensive search being made in order to determine the cause and explain the reason for the physiological reactions due to one's air environment, which have been observed and yet could not be explained in terms of our modern conception of ventilation standards.

It has been noted by the engineers intimately engaged in the health aspect of ventilation problems and by men in the medical field that when the physiological reaction of persons is studied with reference to long-time exposures to certain air environment there is an unexplainable impairment of one's health, and so it has come to many of us that there is need of a more intimate knowledge of facts established by experimentation regarding the less apparent phenomenon, regardless of whether the information leads to a positive or negative finding. It seems reasonable that before a conclusion can be reached respecting the effects of even minute quantities upon our well being, when exposed to these conditions for a long period of time, actual experiments should be made and not rely entirely upon logic.

Physiological reactions can invariably be cited which have been due to exceedingly small quantities of matter. It is well understood, for example, that the secretions of certain glands in small quantities have a profound physiological effect; also, that in the process of immunization only a small quantity of serum will ofttimes

produce lifetime immunity. Even in connection with the respiratory tract, minute quantities of matter have had a noticeable effect on one's reaction. In this instance, I have in mind the presence of inappreciable quantities of substances, the presence of which could be established only with difficulty, although they gave marked stimulation to the olfactory centers.

The author has pointed out that, where ionization takes place in air, invariably substances are formed in much larger quantities than the ionized content, as, for example, nitrous oxide. It does not necessarily follow that these other substances that are present in quantities, are deleterious. For example, ozone has been produced in concentrations irritating to the mucous membranes, and yet by experimental investigations no harmful effect could be indicated due to the nitrous oxide present.

I believe that we should consider the paper as having broadened our knowledge in the field of physics, particularly with respect to certain numerical relationships, but it does not establish the fact that further experimental studies as undertaken by the Society in the electrical properties of the air including ionization are futile. Let me state in closing that I think it is a mistake for this Society to substitute physics and logic for physiology and experimentation.

R. P. COOK (WRITTEN): The Society is indebted to the author for this very timely paper. The conclusions reached seem to have been determined in a logical manner. The warning relative to the possible endangerment of health in experiments on human beings should be heeded.

However, I cannot agree that the air conditioning engineer should entirely cease all study and consideration of ionization as an important factor in his work. It may be that ions have no physiological effect. We do feel sure, however, that there is something other than temperature, humidity, air motion, and odor which plays an important part in human comfort and we should not abandon any possibility until we have proved beyond doubt that it is negligible.

Practically all of the work on the subject of ionization has been done from a physiological standpoint. The air conditioning engineer has begun to suspect the presence of unknown factors in drying and conditioning processes. I know from experience, for example, that certain products susceptible to charges of static electricity, when dried under identical conditions as regards temperature, humidity, air motion, and method of handling, will vary from day to day and from season to season in the magnitude of the charge introduced in drying. Other peculiar effects have been observed in conditioning processes which apparently may only be explained by some unknown atmospheric influence. We do not know that ionization has anything to do with such problems but until we have more to go on than we have at present, I do not believe that we should abandon research on the subject with the statement that it is a waste of time.

C.-E. A. WINSLOW: It reminds me a good deal of a story that has always been told in medicine of the famous physician, John Hunter, who reproved one of his students for too much theorizing and said, "Don't think, try." I think the curse of medicine has been that it has done too much trying and too little thinking, but it is possible to err on the other side.

I fully recognize the apriori validity of Professor Loeb's arguments. We thought when we began work at the Pierce Laboratory that it seemed inherently improbable that ions would have any effect. We thought we would try it, and we have been doing these "absurd and dangerous" experiments now for over a year, and I think we have ample evidence to show that ions may have definite effects under certain conditions.

As Professor Yaglou has pointed out, we have considerable information about the pharmacological action of various gases. In our work we have been using the large ions of magnesium oxide, and we also know a good deal about the toxic action of magnesium oxide. I can assure you that it is just as unreasonable apriori to

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assume that the amount of magnesium oxide used could have any physiological effect as it is to assume that the ionization could have, but the actual experiments which we have conducted under very carefully controlled conditions show that there is a difference of effect in sensitive human subjects between the positive and negative ions that is very distinct. They have the same amount of magnesium oxide, but when that substance is positively charged there are very definite harmful effects.

We are not sure yet about the beneficial effects of the negative ion, but I think that actual experiment shows that there is some flaw in the application of these physical theories.

I do want to add, however, that I am entirely in accord with Professor Loeb and with Professor Yaglou as to the wholly negative evidence of the importance of ions in practical ventilation. In those experiments that we have been conducting we have used millions of ions as compared with the thousands that are found in any practical process of ventilation. Even with those millions of ions we do not get the effect that I have been speaking of with the ordinary normal subject.

We have a paper in press now showing a wholly negative result with normal individuals, but it is when we study the kind of patient that this process has been used for in general, patients with abnormal blood pressures, and with enormous numbers of ions, that we get the effects I have mentioned.

Our results coincide exactly with Professor Yaglou's in suggesting that at present there is no evidence to prove the idea that one can get physiological effects from any such variations in ion content as might be obtained in practical procedure.

MR. AEBERLY: I know that the engineer, as a result of his general training and practice, does not give much thought to physiological effects. I should like to indicate the significance of minute quantities in terms that he uses more frequently. You are all familiar with the destructive action of certain solutions on pipes. In these problems you have been dealing with pH values. It is well understood in the medical profession that between a range of pH 7.3 and 7.5 of the blood lies the existence of man in a reasonably good state of health. In the terms of hydrogen ion concentration this narrow range may be expressed as the difference between approximately 1/20,000,000 and 1/30,000,000 of a normal solution. Much more basic information in blood chemistry than is available today is necessary, and the respiratory functions especially should be more fully understood before final conclusions are even implied respecting the effects of ionization in ventilation. I would rather refer to these problems as a study of the electrical properties of the air than to refer to them in a limited way as ionization. Originally the Society considered these investigations in this broader sense.

MR. THOMPSON: May I make one more suggestion, and that is that the effect may be cumulative. We know that certain poisons, like lead, have cumulative effects. I know from personal experience that carbon monoxide has a cumulative effect which takes some time to be overcome. In an experiment that we made in an oil refinery, with regard to electrostatic charges on tank trucks and on storage tanks, we found that the theory was that the electrostatic charges were set up by friction, by the action of the tires on the road; but when we came to investigate the matter we found that those electrostatic charges were set up by the condensation of moisture from the exhaust or from any jet of steam that happened to be near the tank truck. That charge would accumulate until there was a spark and an explosion.

G. R. WAIT[33]: I am not a physiologist, and I do not presume to say what the effect would be of inhaling electric charges into the human system. You take charges into the lungs with the air you breathe; these charges produce, at the point where they are absorbed, a galvanic current. Such a current might have a decidedly different effect upon the human system than an electric current, for example, arising in consequence of the body being placed in an electric field.

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[33] Dept. of Terrestrial Magnetism, Carnegie Institution of Washington.

I only wish to add this: Every time you walk out in the open air there is an electric current flowing through your body. This is not true of course, if you walk under trees or remain indoors. Because of the earth's electric field, a fairly high current will flow through the body and to some extent to different parts of the body. If such a current has a detrimental effect, then one would expect that those people working on high buildings, and others who are exposed to very intense electric fields (in other words, those who are subjected to intense electric currents) would be more affected than those not so exposed. Statistics do not seem to warrant the conclusion that such people suffer as a consequence of intense electric currents. There may be still other ways in which the human system is affected through the breathing of electric charges that I do not know of. Professor Loeb apparently assumes that the taking of charges into the body, that is, breathing them into the lungs, will have no more effect than an equal number of uncharged molecules, since he compares a given number of charges with a similar number of molecules breathed in. I do not see that this necessarily follows, but it has an important bearing on the subject now being discussed.

There is one other question that one must really consider. When you breathe in ion charges, the chances are they will go into the lungs, perhaps not very far apart, a matter of a few hundredths of a centimeter, but some will be positively charged while others will be negatively charged. Professor Loeb assumes that two charges of opposite sign will cancel their effect. This may not necessarily occur. I do not presume to know whether the two charges will exactly neutralize each other, or whether the small electric current required for their neutralization will have some slight physiological effect.

In another place the author made a calculation as to the possible galvanic current that would be generated through the breathing of charged particles. This is a rather artificial method to follow and I believe it is not necessary to resort to such an artificial illustration. For example, Professor Loeb said, "Let us take one cubic centimeter of solid matter and break it up into particles, 10^{-4} centimeters in diameter, charge it all of one sign and assume that one per cent of it is charged, breathe that in at a certain rate and we will see what current will result." Professor Loeb calculates a current of 10^{-8} amperes. Of course, this whole process is artificial for we do not normally go around breathing lots of finely divided dust charged with one sign.

It might be of interest for me to point out what actually does take place. To indicate the amount of current that is going to result from breathing charged particles actually in the atmosphere, assume that in such a calculation if one breathes in a positive ion and a negative ion they balance; that is, there will be no resulting physiological effect. So the only thing that will bring about a current will be the breathing of more charges of one sign than of the opposite sign. That actually does take place because the earth's field sees to it that there are on an average more positive charges than negative charges in the outside air about us. This results in what we call a space charge. So, we go about in the open air breathing in more positive charges than negative charges which will result in a galvanic current inside the human body, and the magnitude of this current can be calculated.

Taking the result of this space charge, as measured in Washington, D. C., where we have continuous records, we do not get 10^{-8} amperes, as Professor Loeb infers, but only 10^{-16}, which is much smaller. This, as I said before, is assuming that when one takes into the lungs a positive and a negative charge, they neutralize each other's effect, which of course may not take place.

One may also calculate, on an average, the maximum current that would result in Washington, Buffalo, or any other locality. The maximum can be calculated because if you put charges into the air, a space charge results which creates an electric field, hence the more charges you put into the air the greater will be the field until ultimately such a high field is produced that sparking occurs.

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Assume that as many charges are in the air as can exist without sparking taking place from ordinary small elevated objects, a matter of a few feet, and assume this to be the maximum possible field. Then one can calculate the magnitude of the current that would take place in each human system under these circumstances. It comes out, not 10^{-16} as before, but 10^{-11}, which is the highest one should expect in the matter of galvanic currents in the body due to the breathing of charged particles in the open air. Thus we see that such currents are extremely minute but even so, are we mistaken in assuming their physiological effects are nil? It would be more desirable to subject the matter to experimental tests. I can only hope that some organization may see fit to undertake such an investigation to clear up this extremely important question.

W. L. FLEISHER: Professor Winslow, how did they apply the ions which did have effect? I do not think you explained that; the fact that they were not introduced into the general air current.

PROFESSOR WINSLOW: These experiments have been conducted using the machine made by Dessauer in Germany, and the ions are produced by heating a block of especially prepared magnesium oxide which gives off large numbers of ions of both signs. They are then passed through a small grille which removes the desired ions.

This machine is in one room and the air containing either positive or negative ions is fed from a tube through a wall into a small room next door where the patient lies and breathes this air from a mask over the face. The patient does not know whether he is getting positive ions or negative ions, or no ions at all, during the experimental period.

That exposure is continued for an hour or an hour and a half a day. We have been using some of these same subjects for nearly a year, treating them for a month or 6 weeks with one kind of ion, and then for a similar length of time with the other kind, then with the air coming through the machine in the same way but un-ionized.

I think we have completely controlled any psychological factor, and yet it is perfectly clear under those circumstances that while the normal individual shows no effect, the person who has an abnormally high blood pressure shows definitely bad effects from the positive ions. Blood pressure becomes, over that period of a month, progressively higher, and we usually have to stop the exposure because the doctor at the clinic where they are attending regularly says, "What is the matter?" and the patient complains of headaches. When we treat them with negative ions, with the same amount of magnesium oxide for another 6 weeks, those symptoms clear up and the blood pressure goes down.

DR. E. V. HILL: What is the mask for?

PROFESSOR WINSLOW: To get a concentration. It is not a close-fitting mask; it is just a glass cone that goes over the face.

DR. HILL: The entire room is not entirely ionized?

PROFESSOR WINSLOW: There is a considerable concentration all through the room but they are getting millions of ions in the air over their faces—very extreme conditions. We are not quite sure yet whether the relief that is experienced with negative ions is merely due to the absence of the positive ions or not. We are working on comparison of negative ions now. I do think that we have shown that certain susceptible persons definitely respond to the positive charge of the ions.

No. 1006

LARGE-ION AND THE SMALL-ION CONTENT OF AIR IN OCCUPIED ROOMS

By G. R. WAIT * AND O. W. TORRESON ** (*NON-MEMBERS*)
WASHINGTON, D. C.

INCREASING concern in the ion-content of the air, especially that inside of rooms occupied and unoccupied by people, has been shown in recent years on the part of those interested in problems of ventilation and health. In connection with studies of health, K. Egloff [1] in Davos, as reported by K. Kähler,[2] carried out investigations pertaining to the climate inside of rooms. Included with his measurements were those on condensation-nuclei content and small-ion and large-ion content of air, for the most part in unoccupied rooms, but on occasions with people occupying the rooms as well. Diurnal-variation studies of the small-ion and large-ion content of the air of unoccupied rooms were made, as were also tests to determine the effects on the small-ion content of the air during occupancy of the rooms. At about the same time, Yaglou [3] and his co-workers investigated in this country the small-ion content of air inside of unoccupied rooms and compared the diurnal variation with that for outside air-conditions. Yaglou [4] and associates had previously made a study of the effects of occupancy on the small-ion content of air inside of a closed steel chamber. The results of the latter study on the small-ion content of air as affected by the presence of seven people in the closed steel chamber were not confirmed by the results of Egloff when 40 pupils occupied a closed school-room for several hours. The present investigation bears on this point and provides additional information not only on the effect people occupying a closed room have upon the small-ion content but upon the large-ion content as well.

Measurements were made on the small-ion and the large-ion content of the air of a room in the main laboratory of the Department of Terrestrial Magnetism of the Carnegie Institution of Washington; it is believed these may be considered more or less representative of the general office conditions of many individuals. The entire laboratory, including the particular room used for the

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[1] Dissertation, Eidgen. Tech. Hochschule, Zürich, No. 766.

[2] Naturw., 22, 461-463 (1934).

[3] Diurnal and Seasonal Variations in the Small-Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 271.

[4] Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 191.

Presented at the 41st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by G. R. Wait.

measurements, was occupied during the daylight working hours, but was unoccupied in general during the night hours, on Saturday afternoons, and on Sundays. Continuous records of the small-ion and the large-ion content of the air of the room were secured, thus providing data from which mean curves are drawn to represent hour-by-hour variations for the room and building occupied as well as unoccupied. Shortly afterwards, records of the same elements were secured in a room of the Administration Building of the Carnegie Institution of Washington during the annual exhibit of 1933, when large crowds of people were present in the room and building. Still later the apparatus was installed in a room of an isolated building on the grounds of the Department of Terrestrial Magnetism, where special tests were made to ascertain in a quantitative way the extent to which a given number of people in

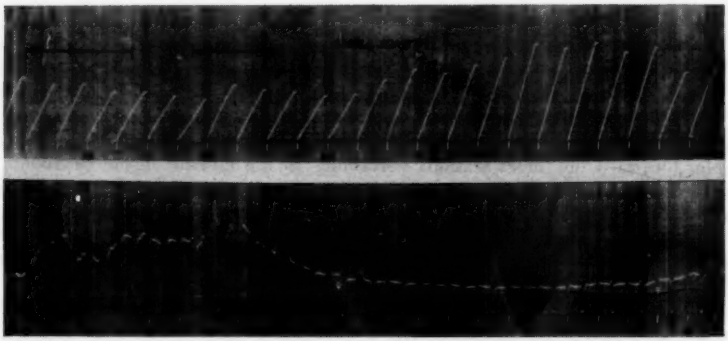

FIG. 1. TYPICAL DAILY RECORD OF SMALL IONS AND LARGE IONS IN ROOM OF ADMINISTRATION BLDG., CARNEGIE INSTITUTION OF WASHINGTON, DURING ANNUAL EXHIBIT, DECEMBER 16-17, 1933 (TIMES NOTED IN 75 DEG WEST MERIDIAN MEAN HOURS—LARGE NUMBER OF PERSONS PRESENT FROM 14.^{h}0 TO 17.^{h}5 AND FROM 19.^{h}5 TO 22.^{h}5 AS INDICATED BY ARROWS)

the room for a given length of time would affect the small-ion and the large-ion content.

The apparatus used has been described in previous publications.[5] The small-ion counter is so placed that the air-stream first enters this instrument before entering the large-ion counter. The large ions, therefore, are counted from the same sample of air as the small ions. Each ion-counter is provided with its own electrometer, the deflections of which provide a measure of the number of ions collected. The central electrode of the small-ion counter and the electrometer to which it is electrically connected continue to charge up over a period of 54 min; the air-stream is then shut off for 6 min, thus providing hourly records for an insulation-test for the small-ion apparatus and for a zero-count of large ions. The charge on the central cylinder and associated electrometer of the large-ion counter is not allowed to accumulate as in the case of the small-ion counter, but is allowed to flow to Earth through a high resistance (about 10^{12} ohms), the resulting *ir*-drop producing a deflection of the electrometer, which is proportional to the number of large ions present. There

[5] *Terr Mag.*, *39*, 47-64 (1934).

is reproduced in Fig. 1 a day's record of the small-ion content and the corresponding record of the large-ion content. A line drawn between the recorded large-ion hourly zero-marks provides the baseline from which to measure the average ordinate. Midway between the hourly zero-marks, there are two dots recorded from automatic calibration of the large-ion electrometer. The small-ion electrometer is calibrated manually at frequent intervals, thus providing ample control of the variables used in the ion-content calculations. The apparatuses have been used to record ion-content of the open air for a long period as well as to record the ion-content of closed rooms.

The closed room at the Department of Terrestrial Magnetism in which the apparatus operated from November 17 to December 7, 1933, being 734 cm by 664 cm and 364 cm high, has a volume of about 179 million cc. This volume was so great that the de-ionizing action of the ion-counters on the air passing through them was not apparent in the results. During regular working hours, the room was occupied by three persons, while there generally were two persons in each of the adjoining rooms. There was occasional smoking of tobacco in the adjoining rooms and in several other rooms of the building but not in the laboratory room. Special tests showed that tobacco smoke affected potently both the large-ion and the small-ion content. All windows and doors were kept closed in the laboratory room to exclude such effects as far as possible. A later test was made to determine whether the effects of occupancy on the small-ion and the large-ion content were due to the presence of the three people in the room or to air-pollution coming into the room from other parts of the building; these will be discussed later.

Records of the small-ion and the large-ion content of the air in the room provide 18 complete days of simultaneous data for study. They have been grouped into four periods, each according to the condition of occupancy of the building and room. All Mondays (building and room occupied from 8 a. m. to 5 p. m.) were placed in the first group, all week-days from Tuesday to Friday (building and room occupied the same as on Monday) were placed in the second group, all Saturdays, including the day before Thanksgiving Day (building and room occupied from 8 a. m. to 1 p. m.) were placed in the third group, and all Sundays (building and room not occupied except for an occasional individual) were placed in a fourth group. The indicated conditions of occupancy were not adhered to strictly and held only for the majority of the time. The week-day group contains six days and each of the others four days of complete record. Mean hourly values of the small-ion content and the large-ion content of the air in the room were obtained for each of the groups. Diurnal-variation curves, beginning with zero-hours Monday and continuing through to midnight Sunday, are shown in Fig. 2. It will be noted that the small-ion content varied in an opposite manner to the large-ion content, both through the day and through the entire period from Monday to Sunday night. The large-ion content was in general high when the building was occupied and low when it was unoccupied. A rapid increase in the content took place almost immediately after the building became occupied and a slow decrease set in from one to several hours after the building was vacated. A gradual accumulation of large ions also took place from Monday morning to Saturday noon, while a slow gradual recovery, in the form of a decrease in the number of large ions, began early Saturday afternoon.

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Tests were made to determine whether the changes in the ion-content of the air of the room were due to occupancy of the room or of the building. During one day the room was not occupied, except for a few minutes at 10 a. m. while attending to the apparatus. Fig. 3 gives comparisons of the diurnal variation of both the large-ion and the small-ion content, while the building, but not the

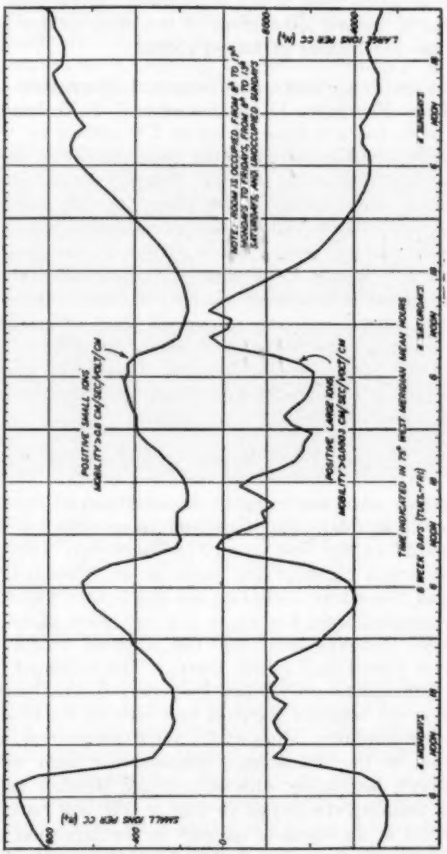
SMALL IONS PER CC (n)
LARGE IONS PER CC (n)
POSITIVE SMALL IONS
POSITIVE LARGE IONS
NOTE: ROOM IS OCCUPIED FROM 8^H TO 17^H MONDAYS TO FRIDAYS, FROM 8^H TO 12^H SATURDAYS, AND UNOCCUPIED SUNDAYS
TIME INDICATED IN 75^H WEST MERIDIAN MEAN HOURS
6 WEEK DAYS (TUES.-FRI.)
MONDAYS
SATURDAYS
SUNDAYS

FIG. 2. EFFECT OF OCCUPANCY ON NUMBERS OF SMALL AND LARGE IONS INSIDE ROOM 210 IN MAIN LABORATORY, DEPT. OF TERRESTRIAL MAGNETISM, WASHINGTON, D. C., NOVEMBER 17 TO DECEMBER 7, 1933

room, was occupied and while both building and room were occupied. The two sets of curves are similar and thus indicate that the principal effects arise from the building being occupied. There is, however, a small consistent difference, the difference in each case being such that it may be attributed to effects arising from occupancy of the room by three persons. The effects from other parts of the building occurred in spite of the fact that all doors and

other openings were kept tightly closed in an effort to isolate the room from other parts of the building. The results emphasize that considerable interchange of air may take place between various parts of a building even with all doors and windows closed. Since the effect on the ions came principally from other parts of the building it was surmised that it was largely due to tobacco smoke. Tests were made to determine to what extent this surmise was correct. Conditions could not be controlled within the main building and special tests were made, therefore, after the apparatus was installed in the isolated building, where all conditions with respect to smoking, occupancy, etc., could be controlled. In this test, smoke from one pipe of tobacco and two cigarettes, during an interval of 22 min, was put into the air of the room and the small-ion and

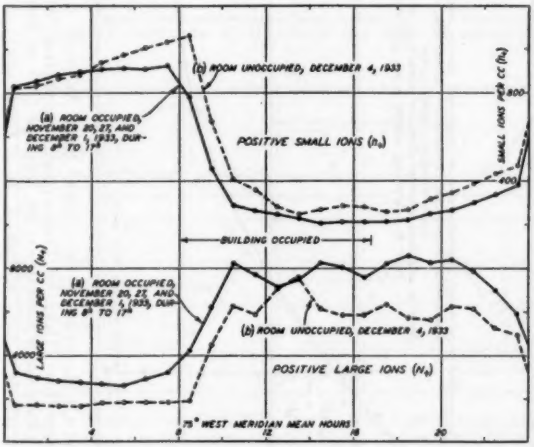

FIG. 3. SMALL AND LARGE IONS, (*a*) BOTH BUILDING AND ROOM OCCUPIED, (*b*) BUILDING OCCUPIED BUT ROOM NOT OCCUPIED, MAIN LABORATORY, DEPT. OF TERRESTRIAL MAGNETISM, WASHINGTON, D. C.

the large-ion content were recorded. The tobacco smoke produced a large increase in the large-ion content and a large decrease in the small-ion content. Both the small-ion content and the large-ion content reversed direction of change soon after smoking ceased, the former gradually increasing and the latter gradually decreasing. This recovery of the two sets of ions continued for more than half a day; the large-ion recovery is shown for one of the tests in Fig. 4. The rate of decrease of large ions ($-dN/dt$) is given by the equation, $-dN/dt = KN$, where K is a constant.

The above results may be explained apparently as follows. Since the smoke particles, when electrically charged, are in effect large ions, the addition of smoke to the air of the room increased the number of large ions; these, combining with the small ions, caused the number of small ions to decrease, more or less corresponding to the increase in the number of large ions. When the smoking was stopped, the charged smoke-particles gradually disappeared from

the air, which in turn allowed the small ions to increase in number. The rate of decrease in the number of charged smoke-particles followed the law $(-dN/dt = KN)$, as expected were they to disappear from the air as a result of falling out under the force of gravity. This being the case, Stokes' law may

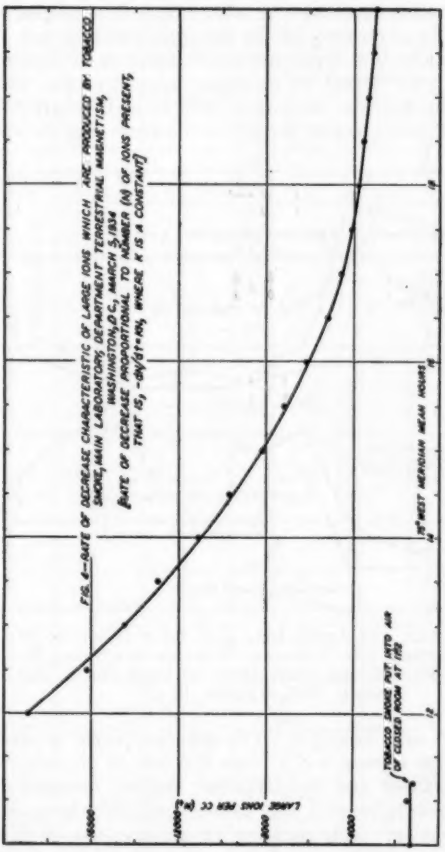
FIG. 4—RATE OF DECREASE CHARACTERISTIC OF LARGE IONS WHICH ARE PRODUCED BY TOBACCO SMOKE, MAIN LABORATORY, DEPARTMENT TERRESTRIAL MAGNETISM, WASHINGTON, D.C., MARCH 5, 1934
RATE OF DECREASE PROPORTIONAL TO NUMBER (N) OF IONS PRESENT; THAT IS, -dN/dt=KN, WHERE K IS A CONSTANT
LARGE IONS PER CC
TOBACCO SMOKE PUT INTO AIR OF CLOSED ROOM AT 11h
10h WEST MERIDIAN MEAN HOURS

FIG. 4. RATE OF DECREASE CHARACTERISTIC OF LARGE IONS WHICH ARE PRODUCED BY TOBACCO SMOKE, MAIN LABORATORY, DEPT. OF TERRESTRIAL MAGNETISM, WASHINGTON, D. C., MARCH 5, 1934 [RATE OF DECREASE PROPORTIONAL TO NUMBER (N) OF IONS PRESENT, THAT IS, $-dN/dt = KN$, WHERE K IS A CONSTANT]

be applied to determine the radius of the particle falling out. The radius of the smoke-particle determined on this basis was found to be almost identical to that determined in a similar manner for the large ions recorded during November 17 to December 7, 1933, in the room of the main building. In other words, the rate of diminution of the large ions in the room after the building was vacated, proceeded at almost the same rate as that for the particles composing tobacco-smoke. This evidence may justify the assumption that the large ions recorded were composed principally of tobacco smoke coming into

the room from other parts of the building. Their effects were so potent that the effects of the people occupying the room itself were small in comparison. Although this conclusion seemed justifiable, it was considered important to compare the radius of a class of larger particles, as calculated on the basis of Stokes' law, with that measured with a high-power microscope. Particles sufficiently large to be seen with a microscope (talcum powder) were injected into the air of the room of the isolated building, and the number of large ions was measured. The rate of decrease followed the same law $(-dN/dt=KN)$ as that for smoke-particles. The radius of particle calculated from Stokes' law agreed almost exactly with that determined with the microscope. It was concluded, therefore, that the assumptions made in the application of Stokes' law were justifiable.

Tests were made to determine if the large-ion and the small-ion content of

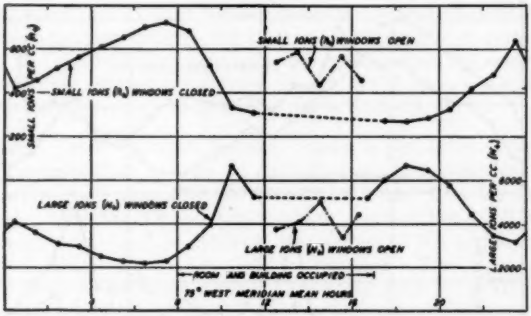

FIG. 5. EFFECT OF OPEN WINDOWS ON LARGE IONS AND SMALL IONS IN OCCUPIED ROOM, MAIN LABORATORY, DEPT. OF TERRESTRIAL MAGNETISM, WASHINGTON, D. C., NOVEMBER 22, 1933

the room were affected through a window being open. Several such tests gave consistent results; typical results are shown by the diurnal-variation graphs of Fig. 5. The effect of the open window is to lower the large-ion content and to raise the small-ion content of the air. The large-ion content, therefore, approaches more nearly that for the outside air, but the small-ion content did not do so; in other words, the small-ion content in the room increased although the small-ion content of the outside air was at the time less than that for the inside air. The small-ion content inside the room is governed, not by the outside conditions, but rather by the large-ion content of the air inside the room, the changes in the small-ion content being in opposite direction to the changes in the large-ion content. The reason for this is that the average life of the small ion is only about a minute or less and sufficient circulation would not ordinarily take place to allow an appreciable number from the outside air to distribute themselves throughout the room.

The tests showed that the small-ion and the large-ion content in the closed

room were noticeably affected when a current of air (produced by an electric fan) was directed from a point several feet below the level of the counters towards the intake of the apparatus. Both the small-ion and the large-ion content showed perceptible decreases when the fan was started, the decreases continuing until the fan was stopped. The effect on the large ions was greatest when the large-ion content was large. The effect, therefore, may have been due to smoke-particles accumulating towards the upper part of the room. The decrease in the small-ion content indicates that the ionization near the floor-level was less than that at other levels.

The negative large-ion and negative small-ion content in a room of the

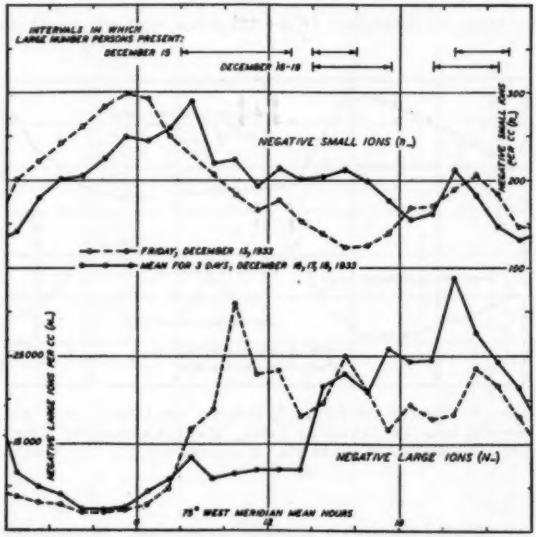

FIG. 6. ION-CONTENT OF ROOM, ADMINISTRATION BLDG., CARNEGIE INSTITUTION OF WASHINGTON, DURING ANNUAL EXHIBIT, DECEMBER 15-18, 1933

Administration Building of the Carnegie Institution of Washington were recorded while the Institution's annual exhibit was in progress during four days in December, 1933. All windows were kept closed but two doors leading to other parts of the building were open during recording. On the first day, December 15, the number of persons present in the building was considerably less than on the three following days. There was occasional smoking in the exhibit-rooms on December 15 but none during the three following days. The hours of the exhibit differed on the first day from those on the three following days; consequently the data have been placed into two corresponding groups for which diurnal-variation graphs are shown in Fig. 6. More-or-less typical

TABLE 1. AVERAGE DEFLECTIONS, SMALL-ION AND LARGE-ION COUNTERS, FOR GIVEN NUMBERS, N, OF LARGE IONS PER CC

	DEFLECTIONS IN DIVISIONS AT CORRESPONDING TIMES					
Large Ions	Large-ion Counter		Small-ion Counter		Differences	
PER CC	D_1	D_2	d_1	d_2	(D_2-D_1)	(d_1-d_2)
4,000	4.0	6.0	25.0	18.3	2.0	6.7
16,000	16.0	18.0	10.0	9.4	2.0	0.6

daily records are reproduced in Fig. 1. While both sets of ions seem to respond in a general way to the effects of occupancy, it is evident that the response of the small ions is less than that of the large ions. Since a reciprocal relationship exists between the numbers of small and large ions in the air, it follows that the small-ion content shows reasonably large responses to moderate changes in the large-ion content, just so long as the large-ion content itself is not great. When the large-ion content is great, the small-ion content changes very little in absolute amount for a given change in the large-ion content. This may be illustrated by average results as given in Table 1.

It is seen from Table 1 that for a given change in deflection of the large-ion counter the corresponding change in deflection of the small-ion counter is much greater for small values of the large-ion content than for large values of this content. Another factor that tends to prevent the small-ion counter from responding to increases in the large-ion content is due to the collection of large and intermediate ions by the small-ion counter. The magnitude of this factor may be large or moderately small, depending upon the particular instrumental arrangements, but at the same time it will depend upon the particular conditions with respect to the number and mobility of large and intermediate ions present. In this investigation this effect has been kept as small as possible, yet permitting the recording of all small ions. From the data of Table 1 it is easy to understand why the small-ion record showed but small response to the effects of occupancy over the interval when the building was occupied, but showed considerable effects later on, in the way of recovery.

The results of Figs. 1 and 6 show that the general effects of occupancy are

TABLE 2. NUMBERS OF LARGE AND SMALL IONS IN ROOM BEFORE AND AFTER OCCUPANCY BY PEOPLE

DATE	PERSONS IN ROOM		SIGN OF IONS		IONS PER CC BEFORE AND AFTER OCCUPANCY				INCREASE IN NUMBER OF LARGE IONS	
					Before		After			
1934	Number	Duration, min.	Large	Small	Large	Small	Large	Small	Per Minute per Person	Per Breath
Jan. 23	10	30	+	−	2,750	445	8,400	269	1.17×10^9	78×10^6
Feb. 15	11	30	−	+	4,070	263	10,170	211	1.14×10^9	76×10^6

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similar in nature to those due to tobacco smoke, as far as the ionic-content of the air of a closed room is concerned. The effects of tobacco smoke, however, are much more potent than is the breath without tobacco smoke. The experiments indicate that the smoking of one individual may alter the large-ion content of a closed room as much as about 100 individuals not smoking. One would expect any alteration in the large-ion content of the air of the room, whether due to tobacco smoke or to other conditions, to produce an alteration in the small-ion content of the air. The results given in Fig. 6 indicate that the small-ion content of the air of an occupied closed room is affected by the presence of large ions. This same effect was noted by Yaglou,[5] who found that the presence of seven people in a closed steel chamber caused a diminution in the small-ion content from about 250 pairs to about 50 pairs in 90 min. Egloff, on the other hand, failed to find any effect on the small ions due to a schoolroom being occupied several hours by 40 pupils. Additional evidence bearing on this point is desirable, therefore. Tests were made in a room of the isolated building, mentioned in connection with another test, with a given number of persons assembled for 30 min in the room. No smoking took place before, during, or after the tests, so that any effects noted with the ion-counters may be ascribed to that due to occupancy. The room measuring 614 cm by 274 cm by 378 cm high contains approximately 62×10^6 cc. The results of the tests are given in Table 2.

The data of Table 2 furnish additional proof that the small-ion content of the air of a closed room is diminished as a result of people occupying the room, as one would expect on the basis of the reciprocal behavior of the small-ion and large-ion numbers. The absolute change in the small-ion content of the air may not be great and, as indicated above, other effects are present which tend to make any decrease less apparent.

The numbers of large ions per breath in the last column of Table 2 are computed from the data on the assumption that they all came from the breath; the total number of both positive and negative large ions is 154×10^6 per breath. From measurements with an Aitken nuclei-counter in a closed room before and after it was occupied, Wait [6] found the number of condensation-nuclei per breath to be, on an average, about 200×10^6. From comparison with the figure obtained for large ions, one would normally conclude that about 75 per cent of the condensation-nuclei are electrically charged. While the respective tests were not made at the same time, the comparison may give a general indication of that portion which comes from the breath electrically charged. This comparison would be quite fair, except that the number of large ions continually decreased with time as shown in Fig. 7. By assuming that the rate of decrease after the room ceased to be occupied occurred also during the time of occupancy, the computed average number of large ions of both signs per breath is about double that given in Table 2, or about 300×10^6. Comparing this figure with the number of condensation-nuclei measured with the nuclei-counter, after making due allowance for their diminution with time, one infers that about three-fourths of the condensation-nuclei were electrically charged. Since the tests of January 23 and of February 15 gave nearly identical values for the number of large ions per breath and since opposite sign of ions

[6] *Jrl. Indust. Hygiene, 16,* 147-159 (1934).

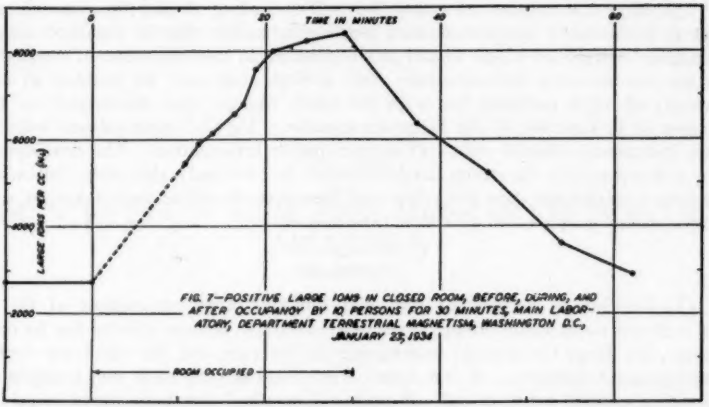

FIG. 7. POSITIVE LARGE IONS IN CLOSED ROOM, BEFORE, DURING, AND AFTER OCCUPANCY BY 10 PERSONS FOR 30 MIN, MAIN LABORATORY, DEPT. OF TERRESTRIAL MAGNETISM, WASHINGTON, D. C., JANUARY 23, 1934

was measured, the indications are that, although most of the condensation-nuclei come from the breath electrically charged, no particular sign predominates. Therefore, it seems probable, within the precision of the measurements made, that electricity is not being removed from the body through the breath particles; additional experiments will be required to determine how far this statement is true.

The rate at which the number of particles in a closed room changes depends upon the difference in their rates of production and of disappearance. Fig. 4 shows the rate of disappearance for smoke-particles in the room chiefly due to their falling out under the force of gravity. Were the number of particles in the room to remain constant, then the rate of production must equal this rate of disappearance. If the rate of production be greater than this limit, then the rate of disappearance must be artificially increased (through ventilation) to keep the number of particles constant until the total rate of disappearance again equals the rate of production. Since the rate of disappearance through falling out is slow, the rate of disappearance through ventilation must closely approach that for production. However, for electrically charged particles of breath the rate of disappearance is rapid (Fig. 7) and hence the rate of production can be correspondingly great if the number of particles may remain constant. For a rate of production greater than this limit the rate of disappearance must be increased through artificial means to keep the number of particles constant. Since, however, the rate of disappearance is normally great, the disappearance through artificial means need not be so great as in the case of smoke-particles, assuming equal rates of production. It may be emphasized that the normal rate of production of smoke-particles by one individual smoking is many times that of breath-particles. Assuming it is desirable to prevent an accumulation of particles (smoke or breath) in a room, it follows that the interchange of air through ventilation must be many times that where there is smoking than that where there is no smoking.

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The large-ion counter arranged for self-recording should find considerable use in problems of ventilation since much information may be obtained with it. Another instrument which should prove valuable in investigations of ventilation is the condensation nuclei-counter since it deals, not with the number of electrically charged particles but with the total, charged and uncharged; to that extent, it is superior to the large-ion counter. Parallel observations with the two instruments should yield still more valuable information. The development of a continuously recording nuclei-counter is extremely desirable, but would require considerable time and effort and therefore should be undertaken by some organization prepared to give that time and effort.

SUMMARY

(1) Continuous records of the small-ion and large-ion content of the air of a closed room made during several weeks show definite effects due to occupancy, the large-ion content undergoing an increase and the small-ion content undergoing a decrease. A few hours after vacating the room and building the large-ion content began to fall off again. A gradual accumulation of large ions occurred during the period of occupancy with a reverse effect in the small-ion content. All these effects were shown to be due principally to the presence of tobacco smoke or other contamination coming into the room from other parts of the building, even though all windows and doors were kept closed except while someone was passing to or from the room.

(2) Continuous records of the small-ion and the large-ion content of the air of a room were secured while many persons were attending an exhibit in the room and in adjacent rooms. As no smoking of tobacco occurred on three days of the exhibit, the effects noted with the ion-counters were due to causes other than to tobacco smoke. No gradual accumulation of large ions occurred in this case, as occurred in the case of large ions from tobacco-smoke. Occupancy, however, did cause an increase in the large-ion content and a decrease in the small-ion content of the air of the room. Recovery of both large-ion content and small-ion content was much more rapid after the building was vacated than when there was smoking of tobacco.

(3) Special tests after the exhibit showed that the small-ion content of the air of a closed room is appreciably decreased as a result of the room being occupied; these results are in agreement with those of Yaglou but in contradiction with those of Egloff.

(4) The special tests indicate that about 300,000,000 charged nuclei are given out in each breath. This, in conjunction with the results by Wait on the number of condensation-nuclei given out in each breath, indicate that most of the nuclei come electrically charged from the lungs. The results further indicate that as many nuclei are charged with one sign as with the other and therefore appear to be removing no electricity from the body.

DISCUSSION

H. S. WALKER[7]: Mr. Wait has done a very good job. He seems to have covered the subject in a well organized manner, and that which he has done is very conclusive.

[7] Detroit Edison Co., Detroit, Mich.

No. 1007

SUN EFFECT AND THE DESIGN OF SOLAR HEATERS

By Harold L. Alt * (*MEMBER*)
New York, N. Y.

IN this paper it is intended to give an analysis of the factors entering into the use of sun heat for warming domestic hot water. The solar heater consists essentially of a coil of pipe exposed to the sun's rays so as to heat the water in the pipe and to circulate the water to a storage tank, in much the same manner as employed in the waterback of a kitchen stove, connected with the kitchen boiler. At first thought it might appear that the water, if heated at all, would only reach a lukewarm state and would not attain a temperature sufficient to serve any practical use. Actually, however, under certain favorable conditions, these heaters serve quite well and have the unique advantage of requiring no fuel for operation.

To date, the design of solar heaters seems to have been one of hit-and-miss, modified by the original ideas of the designer, who may have copied one he has seen somewhere else, or gone ahead blindly to the best of his ability with no guidance at all. The result of such conditions is just what might be expected: some of the solar heaters give good service and others do not. In recent years the advance in the art of heating due to research on heat transfer and sun effect problems makes it possible to apply some of the data which have been developed to the design of solar heaters.

Limitations of Solar Heaters

It must be understood at the outset that solar heaters have certain definite limitations and are not suited for all climates and weather conditions. In fact, they are absolutely confined to those localities where sunshine is stable enough to be counted upon every day in the year, because without sunshine there can be no hot water. They also must be located in climates of temperature sufficiently mild so that the water can be heated to the necessary degree and without the danger of freezing or occurrence of too much radiation from the circulation lines and storage tank. Places where solar heaters are in satisfactory use are principally Florida and the southern part of California. They may, however, be used in almost any portion of the southern United States if augmented by

* Bldg. Equip. Engr., Gibbs & Hill.

Presented at the 41st Annual Meeting of the American Society of Heating and Ventilating Engineers, Buffalo, N. Y., January, 1935.

some auxiliary method of heating, such as a thermostatically controlled gas heater or a thermostatically controlled oil burning boiler with an indirect heater arranged for summer and winter use.

Solar heaters also must be placed in a position where they will receive the sun's rays during the entire day. At no period of time should they be in the shade of any building, or obstruction while the sun is above the horizon. Usually this position is selected on the roof of a building, but it should be low enough to permit the placing of the storage tank at some distance higher than the coil so as to secure a gravity circulation between the coil and the tank. Much better circulation will be obtained if the coil can be placed on a first story roof, or a porch roof, with the tank in the attic one story above, than if the coil is placed on a low portion of the main roof with the tank in the high portion and only a foot or two between the level of the coil and the bottom of the tank.

QUANTITY OF WATER TO BE HEATED

The quantity of water to be heated will be the same as required in any similar building as only the source of heat is changed, but the source of heat is an intermittent one and cannot be counted upon for much more than an approximate average of 8 hours per day. This does not mean that the sun does not shine more than 8 hours per day, but it does mean that it cannot be relied upon to do any very effective heating of water for a much longer period. Therefore, the tank capacity must be sufficient to carry over the other 16 hours when there is little or no sun heating.

If it is assumed that the average person uses about 40 gal of hot water per day, during sixteen twenty-fourths of the day, the consumption based on a straight ratio would amount to 16/24 of 40 gal or about 26 gal. Owing to the fact that a large portion of the 16 hours when no heating is being done occurs at night, also when little hot water is being used, it probably is sufficient to allow only one-half of the quantity of hot water being used during this 16 hour period instead of the two-thirds resulting from a straight ratio. In country locations where people keep early bedtime hours, it might even be slightly less than one-half, but for safety it will be assumed that one-half of the total hot water consumption will occur during the time when the solar heater is not functioning. Therefore the tank must have the capacity to store during the day about 20 gal per person for night use and until the solar heater again goes into operation on the following day.

It is well known that it is impossible to draw the complete amount of stored hot water from a tank without having the incoming cold water cool the remaining hot water down to so low a temperature as to make it unsatisfactory for use, and the amount remaining in the tank when this occurs is approximately 25 per cent. So 25 per cent must be added to the storage capacity to allow the full 20 gal per person to be drawn, making the total storage per person, 20 gal plus 25 per cent of 20 gal or 25 gal. Thus, for a family of four persons a tank with 25 gal $\times$ 4 or 100 gal would be required.

This is graphically illustrated in Fig. 1 where a 100 gal tank is shown with both the amount of usable water stored for the 16 hour period and the amount remaining in the tank indicated.

THE PIPE COIL

In order to receive the greatest amount of heat from the sun, it is evident that the coil should be placed with the pipe surface perpendicular to the sun. As the sun moves daily from the eastern to the western horizons and annually from the farthest north to the farthest south positions, while the pipe coil is fixed in an unchanging position, it becomes evident that the coil must be so placed as to receive the maximum average sun effect for all sun positions. If the coil is placed with one axis running true north and south as shown in Fig. 2, it is evident that it will receive the sun from the east as early, and the sun from the west as late, as any setting that can be devised. This will take care of the daily sun movement from east to west.

In the other direction, allowance must be made for the sun being higher in the sky in summer than it is in the winter and two possibilities suggest themselves. One is to place the coil so the surface will be perpendicular to the sun at noon in its mid-position, as shown in Fig. 3, and the other is to place the coil so that the surface will be perpendicular to the sun at noon in its lowest position. The theory back of the second idea is that a greater amount of heat for the water is required in winter when the sun is at the lowest position and, therefore, that is when the coil should be perpendicular to the noon rays. As the sun comes back north, the outside temperature, water temperature and box temperature all will be higher while the radiation of the tank and circulating pipes will be reduced, thus actually requiring less sun heat to secure the same amount of water warmed to the same degree of temperature. This second position is illustrated in Fig. 4 and has the added advantage of inclining the coil to a greater degree so that air pockets are less liable to form in the coil and so that the coil will be entirely filled with water and work at its highest efficiency.

The point is now reached as to what kind of coil would be best suited for the purpose in mind, remembering that the coil lies flat from east to west but is inclined toward the sun in either the mid position, (see Fig. 3) or the lowest position, (see Fig. 4) as previously noted. This means that the south end of the coil will be lower than the north end, and determines at once that the north end will be the discharge end of the coil while the south end will be the inlet end.

DESIGN OF COIL

The next question is what type of coil will be most desirable and of what material should it be made. Owing to the fact that the coil is to be enclosed in a box, of which more will be said later, it is desirable to have the coil in a square or oblong form; hence a mitre coil is not suited for this work. Neither is a header coil desirable as the water in passing through is heated, and some of the pipes would be at different temperatures from others, resulting in varying amounts of expansion which would tend to open the joints at the header connection. This leaves only the return bend coil to be considered and with such a coil it is readily apparent that the pipes must run from east to west as shown in Fig. 5. If the pipes are run in a direction from north to south, there will be a higher point in each loop at the north end in which air will collect and interfere with the circulation.

If copper tubing with stream line fittings is employed it would seem that the

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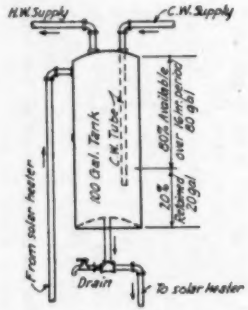

FIG. 1. PROPORTION OF STORAGE TANK CONTENTS AVAILABLE

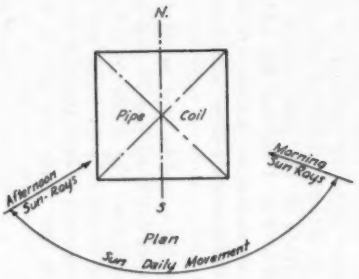

FIG. 2. PROPER POSITION OF COIL WITH AXIS RUNNING NORTH AND SOUTH

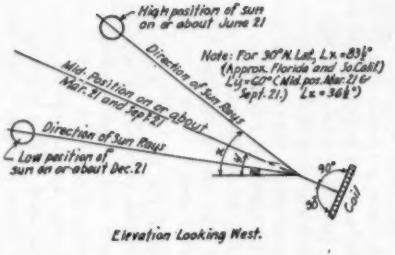
High position of sun on or about June 21
Direction of Sun Rays
Mid. Position on or about Mar. 21 and Sept. 21
Direction of Sun Rays
Low position of sun on or about Dec. 21
Coil
Elevation Looking West.

FIG. 3. COIL WITH SURFACE PERPENDICULAR TO SUN RAYS AT NOON, MARCH 21 AND SEPTEMBER 21

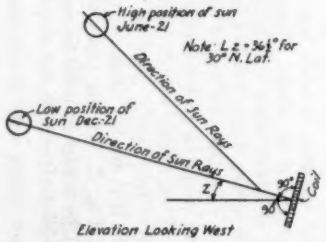

FIG. 4. COIL WITH SURFACE PERPENDICULAR TO SUN RAYS AT NOON, DECEMBER 21

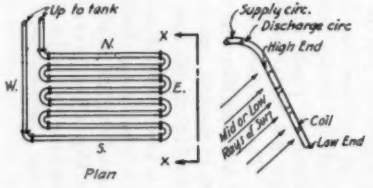

FIG. 5. TYPICAL DESIGN OF RETURN BEND COIL

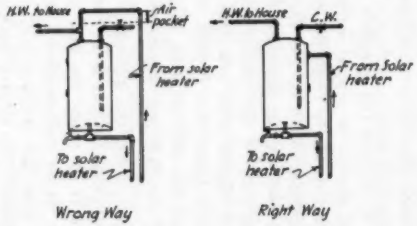

FIG. 6. TANK INSTALLATION TO AVOID AIR POCKETS IN PIPING

highest efficiency would result for the following reasons:

1. Copper has a high heat conductivity which would facilitate the transmission of heat into the water.
2. The stream line fittings permit the flow of water with the least amount of friction and would aid in maintaining circulation with the tank especially at times when the tank temperature approaches the coil temperature and the circulation head is slight.
3. This material is non-corrosive with almost any water, there being only a few cases where copper piping will not stand up.
4. It is light and interposes the minimum weight which can be expected.
5. It will not discolor the water with any rust tint at any time in the future.

Galvanized iron pipe coils with galvanized fittings can be used if desired and possibly may be cheaper in some localities. It is probable that a slight increase in the square feet of coil will be necessary, over that required with copper, if galvanized pipe is used. Whether copper or galvanized iron is used is a matter entirely at the discretion of the designer.

CIRCULATION CONNECTIONS WITH TANK

The circulation connections with the tank should be of the same material as the coil; if a copper coil is used, the same pipe and fittings should be used up to the tank and back and the tank also should be copper. If galvanized iron is employed for the coil, the circulating pipes also should be galvanized to avoid electrolytic action when copper and galvanized iron are incorporated on the same water line and in contact with each other. It is true that the higher heat transmission of copper will result in a slightly greater heat loss on the circulation lines but this will not amount to anything serious especially as these usually are short and should be properly insulated. Air pockets must be carefully guarded against in the circulation lines the same as on any gravity hot water circulation system and it is desirable to connect the coil discharge line to the highest connection on the side of the tank and not to a connection on the top of the tank where an air pocket will be formed—see Fig. 6.

INSULATION OF THE TANK AND LINES

The storage tank should be insulated with some efficient type of insulation, such as hair felt or rock wool, and the insulation should be 1 in. thick. Otherwise there will be quite a heat loss due to radiation from the tank during the 16 hour period when there is no heat supplied by the heater and from the circulation lines during the 8 hours when the heater is in operation. The hot water line from the tank to the fixtures also should be insulated to prevent the drawing off of cooled water whenever a faucet is opened and the wastage of water and heat from this cause. No gravity circulation to fixture outlets can be obtained with this type of hot water supply owing to the height at which the tank must be placed.

CONSTRUCTION OF THE HOT BOX

To secure greater heat transmission to the coil it is customary to locate the coil in a "hot box" which is constructed in most cases of wood with a glass top. This results in the coil receiving heat from two sources; viz., (1) the sun's

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rays and (2) the heated air in the box. The interior of the box should be arranged to reflect all of the heat possible to the coil and to retain in the box all of the heat possible without transmitting the accumulated heat to the outside air. For this reason all of the interior of the box should be painted white in an effort to prevent the sides from absorbing any more heat than necessary and the bottom should be similarly treated. The coil on the other hand should be painted a dull black in order to absorb the heat and transmit it to the water within.

Usually a metallic plate is placed in the bottom of the box with the idea that the sun's rays which do not fall on the pipe will fall on the plate, heat the plate, and that the plate will transfer this heat to the pipe coil by conduction, and by radiation. It would seem that there might be a slight gain in the use of the plate if it were closely bonded into contact with the metal of the coil so as to transmit the heat through the metal rather than via the air.[1] Possibly copper piping with extended fin surface would do this much better but, so far as known, this never has been used.

COIL EFFICIENCY AND AMOUNT OF SURFACE

The amount of coil required is a very much disputed point and this is due no doubt to the different positions, types of coil, materials used and climatic conditions under which solar heaters have been built and operated. According to sun tests made as far north as Detroit,[2] for the middle 8 hours of the day—that is from 8:00 a. m. to 4:00 p. m.—the average Btu received on a hot summer day amounts to about 225 Btu per sq ft, on surfaces in a horizontal plane and without any enclosing box. If it is assumed that in a box 5 ft square with a coil, the coil covers roughly one-third of the area, the other two-thirds being space between pipes and around the sides of the coil, the coil will have a projected area exposed to the sun of ⅓ of 5 ft × 5 ft or 8⅓ sq ft. If the coil were made of 1-in. copper tubing, for example, this would amount to 20 1-in. pipes, 5 ft long (including the return bends) and about 3 in. center to center.

The sun heat on the coil would be

$$8\tfrac{1}{3} \times 225 \text{ Btu per hour or } 1875 \text{ Btu per hour}$$

while the remainder of the heat would be dissipated in the box and would amount to

$$16\tfrac{2}{3} \times 225 \text{ Btu or } 3750 \text{ Btu per hour}$$

As the glass on the top of the box absorbs about one-sixth of the sun's heat[3] only five-sixths of the heat calculated can be assumed as actually entering the box, and, due to shadows from the sides, probably only 75 per cent average of the box is effective during the whole 8 hour period. This reduces the figures for heat actually entering the box to:

---

[1] Copper bottom plates, when used, should be notched out to receive the fittings on the coil in such a way as to allow the coil *pipe* to rest on the plate throughout its entire length and the pipe should be joined to the plate with a continuous solder joint.

[2] THE A. S. H. V. E. GUIDE, 1934, p. 120.

[3] Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford and E. P. Wells, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 285.

For the pipe
1875 Btu × 5/6 × 75 per cent or 1172 Btu per hour.
For the balance of the box
3750 Btu × 5/6 × 75 per cent or 2344 Btu per hour.

To make the balance of the calculations, certain assumptions will have to be made and the results checked with the assumptions to see how far off the assumptions may be. It will be assumed, for example, that the outside temperature is 70 F and that the water enters the coil also at 70 F and leaves at about 120 F or a 50 F temperature rise. Then the average water temperature passing through the coil is:

$$70\text{ F} + \frac{120\text{ F} - 70\text{ F}}{2} \text{ or } 95\text{ F}$$

and, if the pipe absorbs [4] 3 Btu per square foot of external surface per hour for every degree difference between the temperature in the box and the average water temperature, then the heat absorbed by the pipe per square foot will be:

3 Btu × average temperature difference.

The coil as previously determined has 20 pipes 1-in. o.d. and each pipe is 5 ft long so the total coil surface is:

$$\frac{(1\text{ in.} \times 3.14)}{12\text{ in.}} \times 5 \times 20 \text{ ft or } 26 \text{ sq ft plus.}$$

Therefore, the heat absorption for the coil from the air in the box will be:

3Btu × 26 sq ft or 78 Btu, per degree temperature difference.

Now the glass over the top of the box has an area of about 25 sq ft, and transmits about 1.1+ Btu per square foot, per degree difference between the temperature inside of the box and the temperature outside, so that the glass transmission in this case for each degree of temperature difference will amount to:

1.1 Btu × 25 sq ft or 28 Btu.

Consequently, for every degree that the box temperature rises above the outside temperature, there will be a loss of heat from the box amounting to 28 Btu and an absorption of heat by the piping in the box of 78 Btu additional. The absorption of heat by the pipe, however, will not begin until after the box temperature *exceeds* the average water temperature of 95 F for the water passing through the coil because, unless the box temperature exceeds the average water temperature, the heat picked up by the coil when the water is *below* the box temperature will be given off by the coil again as soon as the water is heated *above* the box temperature.

These conditions are plotted graphically in the chart shown in Fig. 7 where an arbitrarily assumed temperature rise of 50 F in the box is used to determine

[4] Transmission through ferrous pipe, water to air generally is taken at 2 Btu per sq ft, per degree difference per hour but the transmission of copper is roughly as 5:3 making the coefficient for copper 3.33+.

the lines for glass transmission and pipe absorption. With 50 F temperature rise in the box the glass will transmit:

$$(120\ F - 70\ F) \times 1.1\ Btu \times 25\ sq\ ft, \text{ or } 1375\ Btu$$

and the pipe will absorb:

$$(120\ F - 95\ F) \times 3\ Btu \times 26\ sq\ ft, \text{ or } 1950\ Btu.$$

These two points have been plotted on the chart above 50 F rise in the box and the glass transmission line has been drawn to 0 Btu for 0 F rise in the box while the pipe absorption line has been drawn to 0 Btu for 25 F rise in

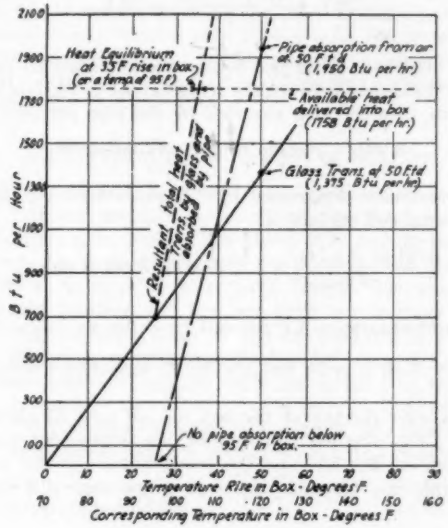
Heat Equilibrium at 35 F rise in box (or a temp of 95 F)
Pipe absorption from air at 50 F t d (1,950 Btu per hr)
Available heat delivered into box (1758 Btu per hr)
Glass Trans at 50 F td (1,375 Btu per hr)
Equivalent total heat transmitted by glass and by pipe
No pipe absorption below 95 F in box.
B t u per Hour
Temperature Rise in Box - Degrees F.
Corresponding Temperature in Box - Degrees F.

FIG. 7. GRAPHIC CHART SHOWING WHERE HEAT EQUILIBRIUM OCCURS

the box which corresponds to a box temperature of 95 F below which point all absorption of heat by the pipe from the box air ceases.

The total heat transmitted into the box under the conditions assumed has been calculated to be about 2344 Btu, which if 25 per cent is assumed as lost in the box, gives about 1758 effective Btu and this line has been drawn across the chart at 1758 Btu on the left hand scale. It is evident that, *when* the glass loss plus the pipe absorption equals the total effective heat developed in the box, no further temperature rise in the box can be expected. Therefore, if the vertical heights for each of the lines for glass transmission and pipe absorption are added together and a resultant total heat utilized line produced, the point at which this total heat line crosses the horizontal line of effective heat available will be the box temperature which will give a condition of heat

equilibrium. This is found to occur at 35 F rise in the box or a box temperature of 105 F.

The heat received from the sun's rays and the heat received from the air in the box may now be combined and the total heat picked up by the coil calculated, this being as follows:

Heat from sun, previously calculated	1172 Btu
Heat from surrounding air (105 F − 95 F) × 3 Btu × 26 sq ft	780 Btu
Total heat received by the coil	1952 Btu

The pounds of water per hour which will be heated 50 F with this amount of heat entering the coil will be

1952 Btu/50 Btu or 39 lb, or
39 lb/8⅓ lb, which is 4.7 gal per hour

and in 8 hours this will give

4.7 gal × 8 or 37.6 gal heated during sunlight time.

As there were 26 sq ft of pipe surface in the box, this shows

37.6 gal/26 sq ft, or 1.4 gal per sq ft per day

as the heating capacity of a solar coil under the conditions assumed.

The practical result of all of this investigation has been embodied in the two schedules shown in Table 1 and Table 2; Table 1 is based on the average use of 40 gal of hot water per day per person while Table 2 has the same information modified to suit the rate of 30 gal of hot water per day per person. The designer must select his own basis for his installation although it would seem that the 30 gal rate in most cases would be adequate.

Some criticism of Tables 1 and 2 may be offered on the following grounds:

(a) The amounts of heat received, area of coil, etc., have been calculated on the basis of the sun effect according to tests in Detroit and the sun effect in the southern part of the country would be considerably greater.
(b) The tests even at Detroit were made on a flat surface which is not perpendicular to the sun's rays at any time, even at noon, whereas (with an inclined pipe coil) the projected surface of the coil would be perpendicular with the sun's rays at noon and therefore would receive more heat than indicated by the Detroit tests.
(c) The water used per occupant may be less than that suggested by the two tables, one for 40 gal per day, and the other for 30 gal per day.
(d) The use of double glass on the top of the box would add to the box temperature due to the reduced transmission to the outside air through the glass.
(e) Practical installations are operating with less surface than shown as required by the tables.

To answer these objections categorically, let the question be considered as to what amount of sun heat will be received in latitudes where solar heaters are in use. Reference to Fig. 8 shows the relation of the earth to the sun's rays in both winter and summer for the northern hemisphere where it will be noted that the sun's rays fall vertically on the Tropic of Cancer when the sun is farthest north, as the saying is, and vertically on the Tropic of Capricorn

when the sun is farthest south. The farthest north position is reached on, or about, June 21 and the farthest south position on, or about, Dec. 21. The time of spring and fall equinox occurs when the sun's rays are vertical over the equator which is on, or about, Mar. 21 and Sept. 21. It will be noted that, in the summer period in the northern hemisphere at the time when the sun is perpendicular over the Tropic of Cancer, the sun's rays are tangent 23½ deg past the north pole—or at the far side of the Arctic circle, while in the winter period when the sun is farthest south, the rays are tangent 23½ deg from the north pole, or at the near side of the Arctic circle.

In order to find the angle with the horizontal at which the sun's rays strike

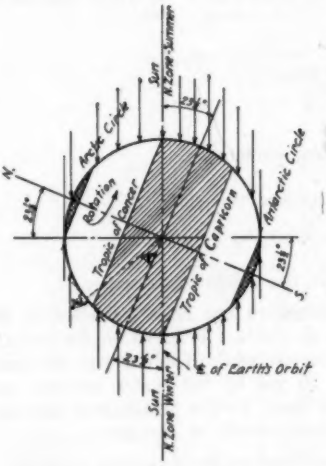

FIG. 8. RELATION OF EARTH TO SUN'S RAYS IN WINTER AND IN SUMMER

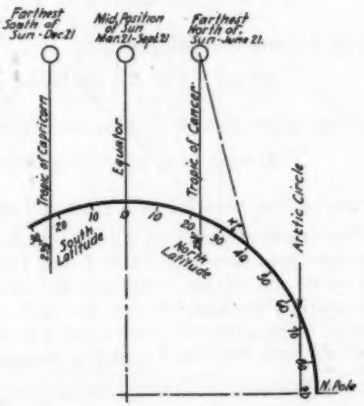

FIG. 8A. DIAGRAM OF ANGLE OF SUN'S RAYS FOR DIFFERENT LATITUDES

the earth at noon on any particular date the following formula may be used:

$$\angle x = 90 \text{ deg} - (\text{Lat. of Place} - \text{Lat. of Sun})$$

which formula applies for all conditions when the sun is north of the equator but which must be modified when the sun is south of the equator to:

$$\angle x = 90 \text{ deg} - (\text{Lat. of Place} + \text{Lat. of Sun})$$

when $\angle x$ is the angle of the sun's rays with the horizontal earth's surface.

Lat. of Place equals the latitude of the place where the angle is desired to be found.

Lat. of Sun is the number of degrees that the sun is north or south of the equator as the case may be.

Fig. 8A shows a more detailed diagram of how this formula may be applied. In the first place it will be assumed that the 40th parallel of north latitude covers a typical portion of the earth about 1000 miles wide and Detroit will

be found to lie within this area. Therefore, the angle of the sun's rays at Detroit on the 21st of June when the sun is farthest north will be by the formula

$$\angle x = 90 \text{ deg} - (40 \text{ deg} - 23\tfrac{1}{2} \text{ deg})$$
or
$$\angle x = 90 \text{ deg} - 16\tfrac{1}{2} \text{ deg which is } 73\tfrac{1}{2} \text{ deg.}$$

Similarly, when the sun is in the farthest south position, or directly over the

TABLE 1. DESIGN DATA BASED ON A HOT WATER USAGE OF 40 GAL PER DAY PER PERSON

ITEM OF DESIGN	NUMBER OF OCCUPANTS IN RESIDENCE							
(Based on 40 Gal Rate).....	1	2	3	4	5	6	7	8
	GALLONS							
Hot Water Used at Night, per Person	20	20	20	20	20	20	20	20
Hot Water Used at Night, Total....	20	40	60	80	100	120	140	160
Retained in Tank, 25%.....	5	10	15	20	25	30	35	40
Tank Capacity Required.....	25	50	75	100	125	150	175	200
Hot Water Used During Day.....	20	40	60	80	100	120	140	160
Total Water to Be Heated in 8-Hour Period.....	45	90	135	180	225	270	315	360
Water to Be Heated per Hour.....	6	12	17	23	28	34	39	45
Copper Coil Surface Required in Sq Ft.....	32	64	96	128	160	192	224	256
Equivalent Length in Feet of 1 in., Copper Pipe.....	128	256	384	512	640	768	896	1024
Box Area in Sq Ft.....	32	64	96	128	160	192	224	256
Size of Box, Width, Ft.....	4	6	8	9	10	11	12	12
Length of Box, Ft.....	8	10	12	14	16	18	19	21

Tropic of Capricorn, the angle x will be:

$$\angle x = 90 \text{ deg} - (40 \text{ deg} + 23\tfrac{1}{2} \text{ deg})$$
or
$$\angle x = 90 \text{ deg} - 63\tfrac{1}{2} \text{ deg or } 26\tfrac{1}{2} \text{ deg.}$$

Now the Detroit test was made on Aug. 1 or 41 days after the sun had reached the farthest north position and, as the sun moves from the Tropic of Cancer to the Tropic of Capricorn in half a year—or 182½ days—the average movement in degrees per day is

47 deg / 182½ days or 0.26 deg per day.

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In 41 days it had moved

$$41 \text{ days} \times 0.26 \text{ deg or } 10.66 \text{ deg, say } 10\tfrac{1}{2} \text{ deg.}$$

Then the latitude of the sun north of the equator at the time of the test must have been approximately

$$23\tfrac{1}{2} \text{ deg} - 10\tfrac{1}{2} \text{ deg or } 13 \text{ deg}$$

and the angle with the horizontal must have been

$$90 \text{ deg} - (40 \text{ deg} - 13 \text{ deg}) \text{ or } 63 \text{ deg.}$$

TABLE 2. DESIGN DATA BASED ON A HOT WATER USAGE OF 30 GAL PER DAY PER PERSON

ITEM OF DESIGN	NUMBER OF OCCUPANTS IN RESIDENCE							
(Based on 30 Gal Rate).....	1	2	3	4	5	6	7	8
	GALLONS							
Hot Water Used at Night, per Person.....	15	15	15	15	15	15	15	15
Hot Water Used at Night, Total.....	15	30	45	60	75	90	105	120
Retained in Tank, 25%.....	4	8	11	15	19	23	27	30
Tank Capacity Required.....	20	40	59	75	94	113	130	150
Hot Water Used During Day..	15	30	45	60	75	90	105	120
Total Water to be Heated in 8-Hour Period.....	35	70	104	135	169	203	235	270
Water to Be Heated per Hour.	4½	9	13	17	21	26	29	34
Copper Coil Surface Required in Sq Ft.....	25	50	75	100	121	145	168	192
Equivalent Length in Feet of 1 in., Copper Pipe.....	100	200	300	400	484	580	664	768
Box Area in Sq Ft.....	25	50	75	100	121	145	168	192
Size of Box								
Width in Feet.....	4	6	7	8	9	10	10	11
Length in Feet.....	6	8	11	12½	13½	14½	16½	17½

This is graphically pictured in Fig. 9. It also may be proved that, if the number of sun rays falling vertically on a square foot of surface is considered as unity, the proportional number of sun rays falling on the same square foot at an angle will vary as the sine of the $\angle x$. See Fig. 10. Therefore, the number of sun rays falling on a horizontal surface at any angle (and, presumably, the sun effect) varies as the sine of the angle at which the rays impinge.

If it is assumed that the 30th parallel of north latitude covers a belt extending approximately 500 miles on each side, it will be found that the major portions of Florida and southern California come within this belt and constitute the sections where solar heaters are in use in this country. The angle at which

these rays fall, when the sun is farthest north, when it is in mid position, and when it is farthest south, then will be by the previously given formula:

$$\angle x = 90 \text{ deg} - (30 - 23\tfrac{1}{2}) \text{ or } 83\tfrac{1}{2} \text{ deg}$$
$$\angle x = 90 \text{ deg} - (30 - 0) \text{ or } 60 \text{ deg}$$
$$\angle x = 90 \text{ deg} - (30 + 23\tfrac{1}{2}) \text{ or } 36\tfrac{1}{2} \text{ deg.}$$

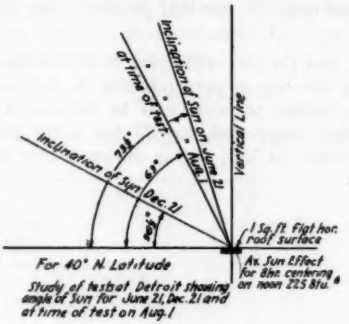
Inclination of Sun on June 21
at time of Test
Vertical Line
Inclination of Sun
73½°
63°
Dec. 21
90°
1 Sq. ft. Flat hor. roof surface
For 40° N. Latitude
Ax. Sun Effect for 8hr. centering on noon 225 Btu.
Study of test at Detroit showing angle of Sun for June 21, Dec. 21 and at time of test on Aug. 1

FIG. 9. ANGLE OF SUN'S RAYS AT TIME OF DETROIT TEST

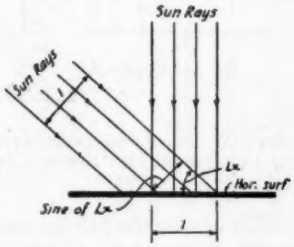

FIG. 10. PROPORTION OF SUN'S RAYS STRIKING A HORIZONTAL SURFACE

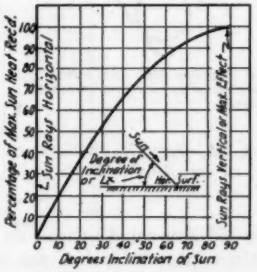
Percentage of Max. Sun Heat Recd.
L. Sun Rays Horizontal
Sun Rays
Degree of Inclination of Lx.
Hor. Surf.
Sun Rays Vertical or Max. Effect
0 10 20 30 40 50 60 70 80 90
Degrees Inclination of Sun

FIG. 11. CURVE FOR MAXIMUM SUN EFFECT AT DIFFERENT ANGLES

With the sun effect as equal to the sine of the impinging angle, and with vertical sun rays as unity, and the angle at Detroit on Aug. 1st as equivalent to an average sun effect of 225 Btu for the 8 hour period centering on 12 o'clock noon, the following table may be developed:

Sin 90	deg, 1.0000	equivalent to 253	Btu average
Sin 83½	deg, 0.99357	equivalent to 250	Btu average
Sin 73½	deg, 0.95882	equivalent to 240	Btu average
Sin 63	deg, 0.89101	equivalent to 225	Btu average (Test)
Sin 60	deg, 0.86603	equivalent to 220	Btu average
Sin 36½	deg, 0.59482	equivalent to 150	Btu average
Sin 26½	deg, 0.44620	equivalent to 113	Btu average.

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Based on the sine equivalent of sun effect, a curve may be plotted showing the percentage of the maximum heat sun effect which will be received for any inclination of the sun's rays as indicated in Fig. 11. Owing to the rounded surface of the pipe, it is evident that, when the coil pipes are run in an east and west direction, the projected surface of the coil is always perpendicular to the sun's rays at noon and the coil itself will receive the maximum sun effect regardless of the distance the sun may be above the southern horizon at noon. See Fig. 12.

It also is evident that the box will not receive the maximum sun effect except when the bottom of the box is perpendicular to the sun's rays at noon; if the box is set with the bottom perpendicular to the sun rays at their *lowest* angle as has been previously suggested, for 30 deg north latitude this will mean an angle with the horizontal of 53½ deg as shown in Fig. 13. Under this condition

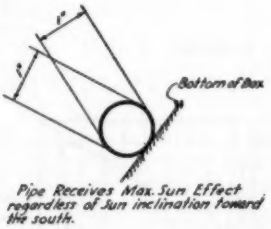

FIG. 12. SUN RAYS FALLING ON ROUND PIPE

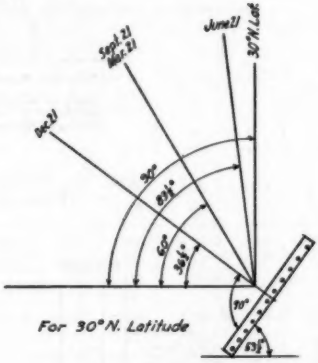

FIG. 13. SUGGESTED INCLINATION OF COIL FOR 30 DEG NORTH LATITUDE

possibly 253 Btu could be used as the sun effect against the 225 Btu used in the calculations presented, but the lower outside temperature—probably around freezing or slightly above—prevailing at such times would increase the glass transmission to a point where the use of this higher figure seems unwarranted.

USE OF DOUBLE GLASS

The use of double glass might be of benefit in cold outside temperatures but if one thickness of glass results in the absorption of about one-sixth of the sun heat it is fair to assume that two thicknesses would have about twice this absorption, which conclusion is borne out by the tests on double plate glass given in THE A. S. H. V. E. GUIDE, 1934.[5] Let the 5 ft × 5 ft box with 100 lineal feet of 1-in. pipe previously considered now be taken with double glass. This gives for the heat effect on the coil

$$1875 \text{ Btu} \times \tfrac{2}{3} \times 75 \text{ per cent or } 937 \text{ Btu per hour}$$

[5] THE A. S. H. V. E. GUIDE, 1934, p. 119.

and for the balance of the box

3750 Btu × ⅔ × 75 per cent or 1875 Btu
or, with 25 per cent lost in the box, 1407 Btu available.

The pipe will still absorb 3 Btu for every degree that the box rises above the average water temperature of 95 F but the heat transmitted out of the box through the double glass will be reduced from 1.1 Btu per square foot, per degree difference, to about 0.45 and this will make the heat loss through glass transmission for every degree rise in the box

25 sq ft × 0.45 or 11¼ Btu.

Replotting Fig. 7 on the basis of double glass and in the same manner as

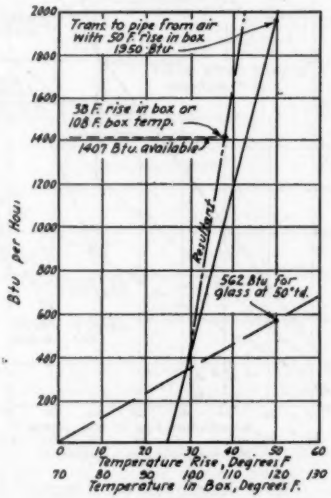

FIG. 14. FIG. 7 REPLOTTED FOR DOUBLE GLASS

previously pursued, it is found that the pipe transmission for 50 F rise in the box remains the same or 1950 Btu (which is to be expected) and that the glass transmission only reaches 562 Btu under the same rise in the box. The resultant of the two lines crosses the line of total heat available in the box at 38 F or when the box reaches a temperature of 108 F which is only 3 F higher than with single glass. See Fig. 14.

For lower outside temperatures double thicknesses of glass do, however, render some real service as is shown in the comparative curves given in Fig. 15 where the same curves are plotted for 35 F outside and which indicate that the box temperature with single glass would only reach 97 F or 2 F above the average water temperature so that the transmission from the air to the coil would practically be lost under this condition whereas, if double glass were

used, the box would reach 109 F in temperature, or slightly better than in Fig. 7 for 70 F outside and with single glass.

The quantity of water to be supplied per occupant must remain a matter to suit the judgment of the designer but if 20 gal per day is desired, the Occupants in Table 1 may be considered as doubled and if 15 gal per day is desired the Occupants in Table 2 may be considered as doubled. It has been the aim in this study more to determine the features necessary to supply a given number of gallons of heated water per day under the conditions obtaining with the solar heater than it has been to determine the amount of water which will be

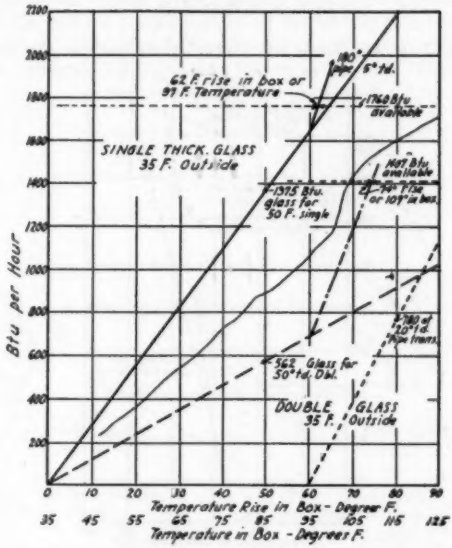
62 F. rise in box or 97 F. Temperature
180° Abs. 5° td.
1760 Btu. available
SINGLE THICK. GLASS 35 F. Outside
Net Btu. available
1375 Btu. glass for 30 F. single
62 F. rise or 107° in box
780 at 20° td. thru trans.
562 Glass for 30° td. Dbl.
DOUBLE GLASS 35 F. Outside
Btu per Hour
Temperature Rise in Box - Degrees F.
Temperature in Box - Degrees F.

FIG. 15. FIG. 7 REPLOTTED FOR DOUBLE GLASS WITH 35 F OUTSIDE

required per occupant as this will vary with the type of person and his status of living.

COMBINATIONS WITH AUXILIARY HEATERS

It might seem feasible to use solar heaters in many localities, where the sun cannot be depended upon at all times, in conjunction with some standby means of heating such as a gas water heater or oil fired steam boiler; by this means the sun heat could be used as long as available and the fuel heater would only come on when the sun heat fails or is exceeded by the load. This should be automatic in arrangement with a thermostat in the storage tank turning on the fuel operated heater only when required. Two typical schemes of such an installation are shown in Fig. 16 where branch circulation pipes are carried down to the fuel heater and the heater is turned on and off automatically by the thermostat located about ⅔ the height of the storage tank. Arrange-

ment should be made to drain the solar heater and piping where temperatures below freezing are encountered.

CONCLUSIONS

Several conclusions may be drawn from this study; *first*, that solar heaters, especially when combined with an auxiliary heater, have a much wider application than has been developed to date. *Second*, that they can be made to furnish a considerable supply of heated water at all times of the year when the sun

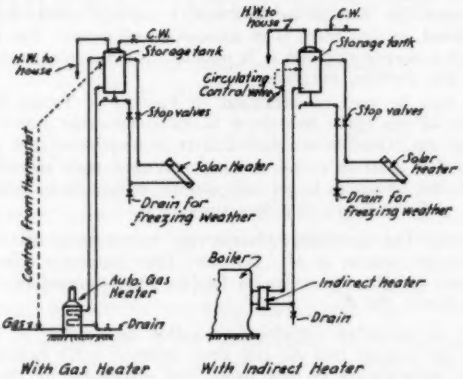

FIG. 16. TYPICAL SOLAR HEATER INSTALLATIONS WITH AUXILIARY MEANS OF HEATING

is available in climates where the outside temperature does not go below freezing. *Third*, that the tank storage should approximate 62 per cent of the daily use to include a 25 per cent retention. *Fourth*, that the box should be inclined toward the sun so that the bottom will be at right angles to the sun rays at the lowest southern position on Dec. 21. *Fifth*, that there is no advantage in placing the coil pipes closer together as the sun effect on the coil will not be increased and the size of box cannot be reduced without the loss of some of the sun heat. *Sixth*, that the coil surface in square feet should be not less than 75 per cent of the number of gallons of water to be heated per day. *Seventh*, that double glass is desirable and beneficial in all cases where the minimum outside temperature goes below 60 F. *Eighth*, that the coil piping and circulation lines should not be less than 1-in. size and 1¼-in. would be even better.

DISCUSSION

L. A. HARDING: I think Mr. Alt should be congratulated on a very practical paper. It is a subject in which people are interested, particularly in the southern states. In so far as the efficiency of absorption is concerned, the best type of solar

heater is a total absorption cone. The plane of the base of cone must be kept perpendicular to the sun's rays. This fact was recognized years ago and it is the principle on which the world's standard pyrheliometer is constructed. Almost any temperature desired can be obtained with a solar heater which is provided with a parabolic reflector to concentrate the rays.

I might add that Mr. Haynes and I visited a residence at St. Petersburg, Fla., equipped with a solar water heater. This solar heater had been in operation for two years and the minimum recorded water temperature, we were told, was 110 F over the two-year period.

THOMAS CHESTER: With reference to the remarks by Mr. Harding, it might interest you to know there are certain isolated astronomical stations on mountain summits where solar heaters are used for cooking purposes. A parabolic mirror is used to concentrate sunlight on a heating receptacle containing fluid which has a high boiling point, so that there is no generation of vapor. The receptacle is usually in the form of a copper coil and it is possible to get a temperature of 500 F which is ample for any cooking purposes.

Solar heat may be used to advantage in Florida or Texas, but for a place like Detroit, in the 42 deg north latitude, it is not comparable with such sources of heat as coal, oil or gas. Due to the fact that it is possible to get 12,000 Btu or more from one pound of coal at very low cost, compared with around 300 Btu per sq ft from sunlight, the economic factor will confine utilization of solar heat to places in low latitudes with relatively cloudless skies.

MR. HARDING: The parabolic reflector type heater referred to by Mr. Chester is Dr. Abbott's solar heater at Mt. Wilson. This apparatus has been employed for cooking, baking and the preserving of foods. The temperature of the circulated oil at times was above 400 F.

H. L. ALT: It is rather surprising to notice the number of text books and reference books on heating that do not even mention solar heating. It was thought that with the presentation of this paper and discussion by the members that the Guide Publication Committee would have enough material to prepare a short chapter for THE GUIDE 1936.

No. 1008

THE THERMO-INTEGRATOR—A NEW INSTRUMENT FOR THE OBSERVATION OF THERMAL INTERCHANGES

By C.-E. A. WINSLOW * AND LEONARD GREENBURG ** (*MEMBERS*)
NEW HAVEN, CONN.

THERE are four distinct external factors which may govern the interchange of heat between the human body and its environment—the temperature of the air and the velocity of movement of the air (governing both convection and evaporation), the relative humidity of the air (governing evaporation), and the temperature of surrounding surfaces (governing radiation). Students of air conditioning have for over a century been seeking, and are still seeking, for the simplest and most accurate method of measuring these four factors and their combined effect upon human comfort.

The classic instrument in use is the thermometer but, while this instrument registers one of the four factors accurately, it gives no measure of either air movement or relative humidity and is affected only in slight and variable degree by radiation. The wet bulb thermometer brings a second factor, relative humidity, within the scope of observation and the Radif thermometer of Lindsay (1933)[1] gives us some measure of radiation, although a very imperfect one, on account of the size and construction of the instrument.

Over a century ago Heberden (1826)[2] realized the importance of recording the influence of air movement and introduced a new principle into the practice of thermometry. He heated his thermometer to above 120 F and observed the time taken by the mercury level to fall for a given distance, under the cooling influence of the air. This is, of course, the principle later developed by Hill (1913),[3] Hill, Griffith and Flack (1916),[4] Hill (1919),[5] Hill (1923 *a* and *b*),[6, 7]

* Director, John B. Pierce Laboratory of Hygiene.
** Associate Director, John B. Pierce Laboratory of Hygiene.
[1] Lindsay, H. B. (Note) *Heating, Piping and Air Conditioning, 5,* 311, 1933.
[2] Heberden, W., An Account of the Heat of July, 1825; Together with Some Researches Upon Sensible Cold, *Phil. Trans., Roy. Soc. of London, 1,* 69, 1826.
[3] Hill, L., The Physiology of Open-air Treatment, *Lancet, 184,* 1283, 1913.
[4] Hill, L., Griffith, O. W., and Flack, M., The Measurement of the Rate of Heat-loss at Body Temperature by Convection, Radiation and Evaporation, *Phil. Trans., Roy. Soc., London,* Series B, *207,* 183, 1916.
[5] Hill, L., The Science of Ventilation and Open-air Treatment, part I, Med. Res. Council, Spec. Report Series 32:46, 1919.
[6] Hill, L., The Science of Ventilation and Open-air Treatment, part II, Med. Res. Council, Spec. Report Series 52, 1923-a.
[7] Hill, L., Kata-thermometer in Studies of Body Heat and Efficiency, Med. Res. Council of Great Britain, Special Report Series 73, 48, 1923-b.

Presented at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by C.-E. A. Winslow.

in the construction of the Kata-thermometer, which has proved an admirable instrument for measuring the two factors of air temperature and air movement. By adding a wet bulb the factor of relative humidity was also brought into the picture, and Hill believed that his wet Kata represented better than any other instrument the combined influence of all atmospheric factors upon the human body. Later studies have shown, however, that this is only even approximately true at certain temperatures and the range of the wet Kata is so narrow that it has not proved of very great value in practical ventilation studies.

About 10 years ago a great deal of attention was devoted to the problem of measuring the combined influence of air temperature, air movement and relative humidity by methods which depended on the same principle introduced by Heberden—rate of heat loss from a heated body. Only instead of measuring heat loss by the rate of cooling of a previously heated bulb the devices now introduced, the Comf-thermometer of Hill (1923)[8, 9] and the Comfortimeter of Phelps (New York State Commission, 1923)[10] both depended on a continuous constant supply of heat to the instrument and a record of the temperature resulting from the balance between this heat input and the loss from the surface of the instrument itself. The Hill device measured the air temperature and the air movement while that of Phelps was covered with a wet cloth so as to be affected by relative humidity as well.

At the same time, the Research Laboratory of the A. S. H. V. E. (1923)[11] introduced an interesting new idea into ventilation practice—that of Effective Temperature. This involved no new instruments but merely the use of a single index expressing the combined effect of temperature, air movement and relative humidity upon comfort, as determined empirically by experiments with human beings.

During the past few years this entire subject has entered upon a new phase with a growing recognition of the importance of the fourth factor governing heat loss from the body—radiation. According to the computations of Aldrich (1928)[12] radiation actually accounts for nearly half the total heat loss from the normally clothed body and yet none of the instruments mentioned above measures this factor with any degree of accuracy. In addition to the crude Radif-thermometer mentioned previously, three interesting instruments have been described to record radiation effects. The simplest of these, which takes account of air temperature and radiation only, is the Globe thermometer of Vernon (1923).[13] This is simply an ordinary mercury thermometer enclosed in a 6-in. or 9-in. sphere of blackened copper. The other two new instruments measure air movement as well, since they are maintained at a temperature above that of the atmosphere.

The Coolometer of Weeks (1931)[14] consists of a copper spool about 1 in. in diameter and 2 in. long, wound with small copper and manganin wires to constitute a combined heating cell and resistance thermometer. A thin shell

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[8] Loc. Cit. See Note 6.
[9] Loc. Cit. See Note 7.
[10] New York State Commission on Ventilation, E. P. Dutton Co., New York, N. Y., 1923.
[11] Houghten, F. C. and Yaglou, C. P., Determining Lines of Equal Comfort, A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923, p. 163.
[12] Aldrich, L. B., Study of Body Radiation, *Smithsonian Misc. Coll.*, 81, 82, 1928.
[13] Vernon, H. M., *Proc. Inst. Heating and Ventilating Engineers, 31*, 100, 1932.
[14] Weeks, W. S., A New Instrument for Measuring Cooling Power: The Coolometer, *Jour. Ind. Hyg., 13*, 261, 1931.

is fitted over the spool to serve as a cooling surface, silvered to measure the influence of air temperature and air movement, or blackened to include radiation effects as well. The instrument is maintained at a constant temperature and the current necessary to produce this result gives the measure of cooling power.

A more elaborate instrument operating on the same general principles had been described in England by Dufton (1930)[15] and more fully by the same author two years later (Dufton, 1932).[16] This device, called the Eupatheoscope, is a blackened copper cylinder 22 in. high and 7½ in. in diameter, preliminary studies having indicated that a body of approximately this size was necessary to imitate the relative effects of radiation upon the human body. The surface of the cylinder is maintained at 75 F, by two electric bulbs controlled by a thermostatic element. A portion of the current supplied to the lamps passes through a coil wound round a mercury thermometer and this thermometer furnishes a measure of the heat input necessary to maintain the skin temperature of the cylinder at the desired point. This thermometer reads on an arbitrary scale of so-called equivalent temperature. A simplified form of this instrument is extensively used in England as a device for controlling radiant-heating systems in actual operation.

The Eupatheoscope is an ingenious and valuable instrument for the measurement of cooling effects in ordinary occupied rooms; but its range is rather narrowly limited. It obviously ceases to operate when the environment is so warm as to raise the temperature of the cylinder above 75 F, or so cool as to exceed the power of the lamps to maintain that temperature of 75 F. Willard, Kratz and Fahnestock (1933)[17] have made certain modifications of the instrument, which render it somewhat more flexible.

For the purposes of the studies planned in the John B. Pierce Laboratory the range of the Eupatheoscope was far too narrow, since it was desired to include somewhat extreme conditions of both hot and cold environments. It was decided, therefore, as suggested by Prof. A. V. Hill (1932), to reverse the Dufton principle of maintaining a constant surface temperature and measuring heat input and, instead, to supply a constant heat input and record the surface temperature resulting under various environmental conditions. The instrument devised along these lines, which has been called the Thermo-Integrator, has proved so useful that it seems desirable to describe it at this time.

DESCRIPTION AND MODE OF OPERATION OF THE THERMO-INTEGRATOR

The Thermo-Integrator, Fig. 1, consists of a hollow copper cylinder 24 in. long and 8 in. in diameter with hemispherical ends, the metal being approximately 0.05 in. in thickness. Each end is provided with a cylindrical nipple ½ in. in bore and about 1 in. in length, the nipples being threaded on the outside. Over each end nipple is fitted a brass cap suitably threaded and provided with a central opening. Through one of these brass caps there passes a piece of brass rod about ⅛ in. in diameter while through the other a piece of copper tubing ¼ in. in diameter is inserted. Both the rod and the tube are insulated

[15] Dufton, A. F., The Effective Temperature of a Warmed Room, *Phil. Mag.* 9, (59), 858, 1930.
[16] Dufton, A. F., The Temperature of a Warmed Room. The Eupatheoscope, Department of Scientific Research of Great Britain, Building Research. Technical Paper No. 13, 1932.
[17] Willard, A. C., Kratz, A. P. and Fahnestock, M. K., The Application of the Eupatheoscope for Measuring the Performance of Direct Radiation and Convection in Terms of Equivalent Temperature, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 303.

from the shell by means of rubber stoppers through which they are inserted. A suitable length of No. 24 Nichrome wire serves to join the two inside ends of the brass rod and copper tube. The tube and rod in this way serve as terminals for the Nichrome filament which is the heating source for the instrument. In addition, the cylinder is evacuated to less than 0.1 per cent of an

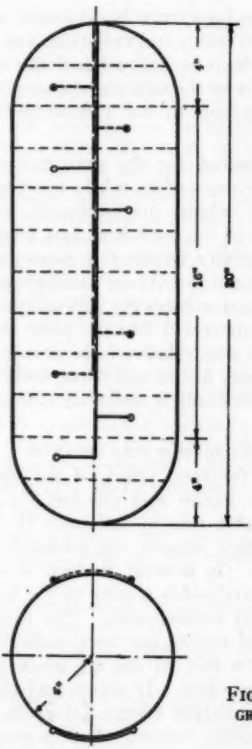
FIG. 1. THE THERMO-INTEGRATOR

atmosphere through the copper tube, a procedure which aids in maintaining a rather uniform surface temperature by minimizing convection currents and air stratification within the instrument. The heater circuit is connected to two 6-volt storage batteries and adjusted to dissipate 21.5 w of energy. This is equivalent to a heat dissipation by the instrument of 17.5 Btu per square foot per hour.

The copper cylinder was made in two parts from two large sheets of copper by the well known metal spinning process, after which the parts were trimmed

to dimensions. The two halves of the cylinder were then soldered together, after which the whole was electroplated so as to make an air-tight vessel. The two nipple ends previously described were turned on a lathe separately and then soldered in place prior to electroplating.

Eight iron-constantan thermocouples (Fig. 2) were distributed over the periphery of the Thermo-Integrator in order to record its surface temperature. The thermocouples were arranged in a parallel circuit and connected to an



Physical Dimensions

Length of Thermo-Integrator	24"
Diameter	8"
Surface Area	603 sq.in. or 4.187 sq. ft.
Mass	5840 grams or 12.43 lbs.
Density (Copper)	8.89 gms/cc.
Specific Heat	0.094

Heating Circuit

Heater Wire	#24 Nichrome
Diameter	0.020"
Length of Heater Wire	60 cms.

FIG. 2. THE THERMO-INTEGRATOR, SHOWING THE LOCATION OF THERMOCOUPLE ELEMENTS

automatic recorder so as to obtain the average surface temperature. In this manner it was easily possible to obtain a continuous record of the average surface temperature of the Thermo-Integrator.

The mode of operating the Thermo-Integrator is exceedingly simple; the steps may be enumerated as follows:

(1) Make proper electrical connections to the filament of the Thermo-Integrator to supply a constant energy input. Two 6-volt storage batteries, floated by a 4-amp

Tungar battery charger were used to supply a constant current of 2.5 amp. The voltage drop across the filament is determined from time to time to insure the proper wattage input to the instrument.

(2) Set the instrument up under the atmospheric conditions the effect of which is to be recorded and allow time (ordinarily about half an hour) for equilibrium to be established.

(3) Determine the surface temperature of the Thermo-Integrator. Make ambient air temperature measurements.

VALUES AND LIMITATIONS OF THE INSTRUMENT

The surface temperature of the Thermo-Integrator is obviously affected by the three factors of air temperature, air movement and radiation, and unaffected by the fourth factor, relative humidity. No one has yet had the temerity to devise an instrument which will record all four factors simultaneously although it might be done by adding a moist covering to the Eupatheoscope or the Thermo-Integrator.

If one of the four factors is to be ignored for the present it seems that relative humidity is the one to neglect. Evaporation, while exerting a major influence at high temperatures, accounts within the ordinarily comfortable temperature range for less than a quarter of the total heat loss from the body and rather wide variations in relative humidity within this temperature range cause relatively slight changes in this proportion. At an atmospheric dry-bulb temperature of 70 F, with 20 per cent relative humidity, the evaporative heat loss from the body of an average man is about 115 Btu per hour, while at the same atmospheric temperature with 95 per cent relative humidity, it is reduced only by 30 Btu per hour (Houghten, Teague, Miller and Yant, 1931).[18] Thus, an enormous change in relative humidity has only a moderate effect upon total heat loss.

It is hoped, of course, that the Thermo-Integrator may serve as a reasonably satisfactory instrument for representing the total effect of the environment, excluding relative humidity, upon human health and comfort. Comparative studies of an elaborate nature are now under way to determine this point. It may be noted, however, that several factors suggest the probability that the instrument may be serviceable in this way. In general, its heat input of about 17 Btu per square foot of surface area corresponds closely to the heat production of the human body per square foot of surface area. The actual surface temperature of the Thermo-Integrator is of course far more variable than that of the human body, being 10-12 deg above that of the air when there is no marked air movement and slight radiation effect. It seems probable that the instrument may tend to over-emphasize radiation effects (since in still air it loses two-thirds of its heat by radiation, while Aldrich's figures would suggest that the human body loses by radiation only three-fifths of the total heat loss minus the evaporative heat loss). It is also very markedly influenced by air movement. How valuable the Thermo-Integrator may be as a general measure of atmospheric comfort, only subsequent studies will determine.

It is certain, however, that this instrument has very great value from another angle—as a means of recording the total radiation effects of a given environmental condition. In an ordinary room, variations in radiation effects, from

[18] Houghten, F. C., Teague, W. W., Miller, W. E. and Yant, W. P., Thermal Exchanges Between the Bodies of Men Working in the Atmospheric Environment, *Amer. Jour. Hyg., 13,* 415, 1931.

windows and walls, from sources of heat and light, and from the bodies of other occupants, are extremely variable and almost impossible to sum up accurately by any method of direct measurement. The Thermo-Integrator automatically summates all such radiation effects and, after calibration, can be used to measure them after subtracting from its total heat loss the losses which would be caused by the temperature and velocity of the air as directly measured under the given conditions. It is largely on account of its value for this purpose that the instrument is described at this time.

DISCUSSION

M. K. FAHNESTOCK: Having used the eupatheoscope which Professor Winslow mentioned, it is true that it is somewhat limited in use, but not quite as much as he has perhaps indicated, because in our tests we used a fixed surface temperature and varied the heat input to the instrument according to the heat loss from it. By changing the heating element in the unit, it is possible to use that instrument at any temperature below the surface temperature at which it is calibrated.

The calibration temperatures are the temperatures at which we measure the heat loss from the instrument in uniform environments, and the instrument which we use has been calibrated for a surface temperature of 75 F and uniform environments, which is the same as the British calibration. For American use we have also calibrated it in uniform environments with the temperature of the surface at 83 F. Therefore by changing our heating element in the unit, we can use it from any temperature from 83 F down to zero, or any practical temperature.

The instrument was developed and used for evaluating non-uniform environments in terms of uniform environments and we thought that it would simulate a human being seated at rest in the environment.

In order to measure the convection and radiation heat loss similar to a human being, we thought that the surface temperature should be nearly the same as the surface temperature of the being or person in the environment. For that reason we held the surface temperature constant.

I would like to ask Professor Winslow if in using the thermo-integrator in a rather cool environment, and having fixed the heat input to it, if the surface temperature does not drop an appreciable amount as the environmental temperature decreases?

With the heat input to the thermo-integrator fixed so that it is dissipating 17.5 Btu per square foot per hour in a uniform environment of 70 F, what would be the change in the surface temperature of the instrument if it were placed in a uniform environment of 60 F?

The instrument is perhaps more flexible than the eupatheoscope which we built and used several years ago.

L. A. HARDING: I think that possibly the instrument might find a practical application in determining whether it is possible in a certain air conditioned room to maintain temperature and relative humidity conditions that will fall within the comfort zone. In a small room you might, for example, maintain a temperature of 78 F dry-bulb and 30 per cent relative humidity, for the sake of argument. This condition lies within the comfort zone and yet you may feel uncomfortable due to the body radiation loss to the cold wall.

C.-E. A. WINSLOW: I agree that the eupatheoscope is a better instrument for the direct measurement of human comfort. We get tremendous differences with this instrument, maybe 20 or 30 F drop in the surface temperature of our instrument. It is really a physical instrument for the measurement of radiation effects over wide

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ranges and we believe it can only be related to human comfort by calibrations and formulae which will correlate it.

The eupatheoscope is excellent for the purpose it was designed. We wanted to get something which would measure radiation effects outdoors, indoors, and under a wide variety of conditions. For that purpose the thermo-integrator is useful. Later we hope to be able to put our observations together, to expand the idea of the comfort zone to include the radiation factor, but first of all to measure that factor.

I heartily agree with Mr. Harding about the difficulty of measuring human comfort. Let me give one illustration. We are working now, as a preliminary, on unclothed subjects in copper booths to get maximum radiation effects. In one experiment we had the subject at an air temperature of about 60 F, and with a powerful radiating influence from the walls. His skin temperature was about 7 F above the normal skin temperature, and yet he felt cold.

No. 1009

A LABORATORY STUDY OF MINIMUM VENTILATION REQUIREMENTS: VENTILATION BOX EXPERIMENTS

By W. H. LEHMBERG,* A. D. BRANDT,** AND KENNETH MORSE *** (*NON-MEMBERS*), BOSTON, MASS.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with The School of Public Health, Harvard University.

THE purpose of this work was to study by laboratory tests the relationship between the intensity of body odor in an occupied space and the quantity of fresh air supplied, the factors affecting odor intensity, and finally the possibility of utilizing odor intensity as an index of minimum ventilation requirements.

It is generally recognized that in crowded and poorly ventilated rooms the air acquires a disagreeable odor which is especially repulsive upon entering from clean outside air. Just what is the characteristic "human smell" is not known. It is believed to be a compound odor arising from emanations of sebaceous secretion, especially when personal hygiene is defective, from decaying teeth, infected tonsils, from the stomach, bowels, skin, etc.

There is considerable difference of opinion whether pollution of room air by such offensive odors should be considered a public health matter. Many health authorities are of the opinion that, while such odors may be offensive, they are not directly poisonous nor are they known to cause any specific and lasting disease. The odor, nevertheless, may be so offensive as to cause reflex digestive and other systemic disturbances which may interfere with growth and general activity. According to the studies of the New York School Committee on Ventilation [1] there is a reduction in appetite and a disinclination for physical work.

Fortunately, or unfortunately, the occupants of a smelly room lose their ability to recognize the odor soon after entering, but this does not necessarily mean a loss of effect. Breathing fresh air restores the ability to recognize the odor anew. To obtain a comparison of odor intensities, therefore, one should always pass directly from fresh air to the odorous air and note the

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[1] Report of the New York State Commission on Ventilation, E. P. Dutton Co., New York, 1923.

Presented at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by W. H. Lehmberg.

odor sensation immediately. For an excellent brief discussion of the anatomy and physiology of the sense of smell, the reader is referred to a recent review by Fair.[2]

Next to the thermal condition of the air as indicated by its temperature and humidity, the intensity of odor observed upon entering an occupied room from outside fresh air furnishes the most valuable evidence of good or bad ventilation. The general agreement is that a certain amount of fresh air from outdoors is necessary at all times in order to dilute the odoriferous matter to a concentration that is not objectionable. The amount needed varies so much under various conditions that it seemed advisable to study the problem first in the laboratory where the various factors can be controlled. The results of the present study are, therefore, limited to the laboratory conditions under which they were obtained.

APPARATUS

The experiments were carried out in the psychrometric laboratory of the Harvard School of Public Health. Male subjects of various body weights and heights were confined, one at a time, in an airtight wooden box, 1 ft 8 in. × 1 ft 8 in. × 6 ft 4 in. long, which was placed inside the psychrometric chamber and connected to the supply and exhaust system of the air conditioning plant. The interior of the box was painted with three coats of waterproof paint. Two windows on the top admitted sufficient light for reading or writing. Clean outdoor air, conditioned and metered, was introduced at the head end of the box through a 4 in. galvanized duct, and distributed uniformly along the length of the box by the use of a U-shaped perforated manifold inverted over the head of the subject. The used air was exhausted through a similar pipe, at the foot end of the box, leading to the exhaust fan of the psychrometric room.

Suitable holes having the shape of the nose were cut in the exhaust pipe for smelling the air coming out of the box.

The ventilation rate through the box was measured by the use of calibrated orifices in the supply pipe. It was controlled by means of a damper and by regulating the speed of an auxiliary direct-current fan unit.

Dry and wet bulb thermometers in the supply and exhaust pipes, close to the box, showed the increase in temperature and moisture content of the ventilating current in passing through the box. In addition the temperature of the air at various points inside the box was recorded in order to obtain the average box temperature.

PROCEDURE

Trained subjects came to the laboratory, one at a time, about 9 a. m. or 1 p. m. and lay quietly and comfortably on a cot or in the open box for a preliminary resting period of one-half to two hours, depending upon previous activity. At the end of this period the subjects were weighed and closed in the box where they lay against the bare wooden bottom with only a small pillow under the head. The desired test conditions being fixed in advance, the subjects remained in the box until the odor intensity, temperature, and humidity of the air coming out of the box reached a steady state. This usually took one to

[2] Fair, G. M.: On the Determination of Odors and Tastes in Water, *Jour. of New England Water Works Ass'n.*, XLVII, Sept., 1933.

two hours depending upon the rate of air flow and temperature. In some experiments the test period was prolonged to four hours or more.

The subjects in the box kept records of their own pulse, body temperature, respiration rate, comfort, and general impression as to air quality. Their spare time was usually taken up by reading. Altogether 10 men, between the ages of 20 and 37 years, served as subjects. They were apparently healthy and had normal personal habits.

Observations of odor intensity were made by 15 trained judges, including smokers and non-smokers, who first smelled the odorless air in the psychrometric room (air circulation through room at least 750 cfm per person all from outdoors) for a few minutes, and then the air coming out of the box, through the openings provided on the side of the exhaust pipe. One to three breaths at the most gave full response to odor. The judges were employees of the school. As many of the 15 as it was possible to get hold of in each experiment

TABLE 1. SENSORY INTENSITY SCALE OF BODY ODOR

ODOR INTENSITY INDEX	CHARACTERISTIC TERM	QUALIFICATION
0	None	No perceptible odor.
½	Threshold	Very faint, barely detectable by trained judges; usually imperceptible to untrained persons.
1	Definite	Readily detectable by all normal persons but not objectionable.
2	Moderate	Neither pleasant nor disagreeable. Little or no objection. Allowable limit in rooms.
3	Strong	Objectionable. Air regarded with disfavor.
4	Very strong	Forcible, disagreeable.
5	Overpowering	Nauseating.

were called in when needed. They voted confidentially according to the scale in Table 1, without knowing particulars about the test conditions. Their records were finally averaged for each test and the mean odor intensity plotted against the corresponding air flow.

The zero base of the scale is, of course, not absolute but it represents the sense of smell of the air in the psychrometric room and in the supply to the box, the latter being a portion of the air stream leading to the psychrometric room and differing from it in temperature only. As the entire air conditioning system, including ducts and registers had been washed thoroughly prior to this study, it would be reasonable to assume that the air in the psychrometric room did not differ appreciably in smell from the air out of doors. An odor strength of zero was not recorded in any box experiment with air flows up to 50 cfm, as compared with the controlled air flow of 750 cfm per person in the psychrometric room.

Odor index No. 2 depended to some extent on the sensitivity or tolerance of

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the judges to smell, but with the exception of two women, the records of the men agreed within ± ½ point or less on the 5 point scale. As a rule the two women judges voted higher than the men. Before training, the judges preferred to record odor strength in whole numbers, but afterwards they felt confident enough to vote to the nearest half division on the scale. Likewise, the threshold intensity index declined from 1 before training to ½ after training.

For full response, the smelling of the odor had to be done in one to three breaths at the most, and the mucous membranes had to be neither too moist nor too dry. Blowing the nose lightly just before smelling helped in all cases.

It was often necessary to wash with soap and rinse out thoroughly the inside surfaces of the box, fan, and exhaust duct, so as to begin all experiments with

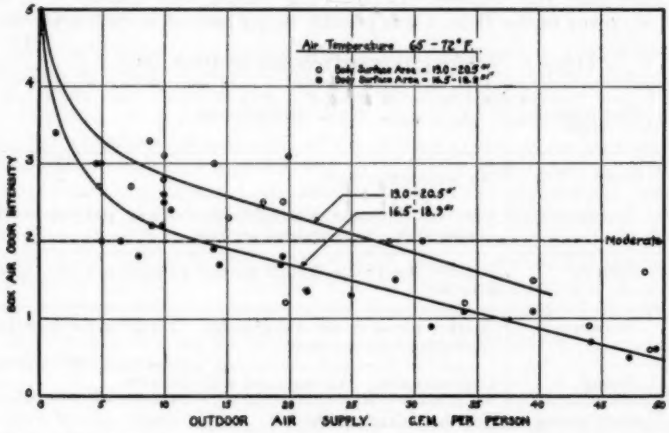

FIG. 1. ODOR INTENSITY IN RELATION TO VENTILATION RATE AND BODY SURFACE AREA

an odorless box. This is an important point because it has a bearing on the claim sometimes made to the effect that odors in occupied rooms cannot be controlled by ventilation. A properly ventilated room never gets to that stage. On the other hand, a poorly ventilated room would require many hours of flushing with clean air to get rid of the odor absorbed by the walls, furnishings, clothing, etc.

In parallel with these observations a series of physiological tests was carried out, the results of which will be presented in another paper.

RESULTS

Odor Strength in Relation to Ventilation Rate and Body Surface Area

In Fig. 1 are shown the results of all experiments with box temperatures between 65 and 72 F. Proceeding from the point of maximum odor intensity 5, which corresponds to air flows under 1 cfm, the odor intensity decreases rapidly as the ventilation rate increases to about 10 cfm. After that the de-

crease in odor intensity is gradual, varying almost inversely as the air flow to the first power. According to Fig. 1, the minimum outdoor air supply per person necessary to prevent the accumulation of objectionable body odor in the air of the box varied from 13 cfm for subjects with body surface areas between 16.5 and 18.9 sq ft, to 27 cfm for surface areas between 19.0 and 20.5 sq ft. The average adult has a surface area of 19.5 sq ft.

From this it would seem that body surface area is a very important factor affecting the amount of odoriferous matter given up to the air. The relationship is now being studied more thoroughly in ordinary rooms using school children for subjects as well as adults.

Complete elimination of odoriferous matter in the air of the box was never attained in any of the experiments with ventilation rates up to 50 cfm. This

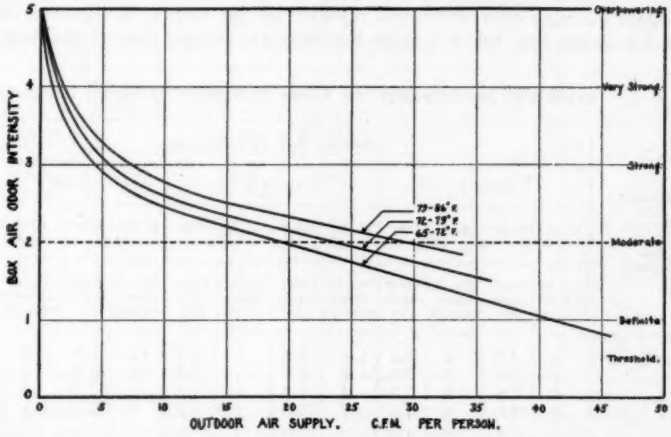

FIG. 2. THE INFLUENCE OF AIR TEMPERATURE ON ODOR INTENSITY

comparatively high air supply was barely sufficient to reduce the odor intensity to the threshold (intensity index ½).

The variability of the results, as shown by the dispersion of individual points from the general trend, is to be expected in tests of this nature. The frame of mind, personal hygiene, type and amount of food eaten, days between last bath and change of underwear to the day of test, proved to be factors influencing the results. Some of these were controlled, more or less, by selecting subjects of normal habits; others by training.

In experiments on a larger scale, such as the ones that are now being carried on in ordinary rooms with as many as 14 subjects at a time, the personal factors seem to be so nicely balanced as to yield very much better results.

The Influence of Temperature on Odor Intensity

In Fig. 2 the results of all experiments are classified in three groups according to the temperature of the air in the box. A rise in temperature is associated with an increase in odor intensity, owing probably to increased perspira-

tion, but the effect is not so great as is generally believed. The minimum ventilation requirements for the three different temperature ranges are about 20 cfm at 65-72 F, 24 cfm at 72-79 F, and 30 cfm at 79-86 F, under the experimental conditions. The last air flow is substantially that demanded by ventilation laws and codes in several states and cities of the U. S. A.

Humidity in Relation to Odor Intensity

Little confirmation could be found for the general belief that high humidities intensify the odor of occupied rooms. As can be seen in Table 2, the difference in odor intensity at relative humidities over 40 per cent (40-90 per cent) and under 40 per cent (15-39 per cent) is too small to deserve mention.

Air Quality in Relation to Air Supply

Despite the high odor intensities reported by the judges, who compared the fresh air going into the box with the used air coming out of the box, the

TABLE 2. INFLUENCE OF HUMIDITY ON ODOR INTENSITY (JUDGES' IMPRESSIONS)

Outdoor Air Supply cfm per Person	Average Box Temperature											
	65–72 F				72–79 F				79–86 F			
	R. H. Under 40%		R. H. Over 40%		R. H. Under 40%		R. H. Over 40%		R. H. Under 40%		R. H. Over 40%	
	No. of Observations	Odor Intensity	No. of Observations	Odor Intensity	No. of Observations	Odor Intensity	No. of Observations	Odor Intensity	No. of Observations	Odor Intensity	No. of Observations	Odor Intensity
Under 5.....	2	3.9	3	3.6	2	3.5	7	3.6	2	4.0	4	4.4
5–15.....	5	2.6	7	2.3	8	2.8	11	2.3	5	2.8	2	3.4
15–25.....	4	1.8	3	2.4	4	1.9	4	3.0	1	2.4	4	2.5
25–35.....	2	1.7	5	1.3	2	1.6	3	1.7	4	2.3	3	1.6
All air flows	13	2.4	18	2.3	16	2.5	25	2.7	12	2.8	13	3.0

subjects inside the box were as a rule unconscious of odor, owing to olfactory adaptation, except for a sensation of staleness, flatness, or lack of freshness in the air as perceived by the sense of smell. Such sensations constituted, in fact, the chief criterion of bad air in the opinion of the subjects. Complaints of this sort were most frequently associated with ventilation rates under 5 cfm even when the air was neither too warm nor too cool. A few of the subjects claimed to have actually smelled a weak body odor throughout the test period with ventilation rates up to about 10 cfm.

Table 3 shows the impressions of the subjects as to air quality under various ventilation rates. Since air quality depends so much upon temperature and humidity, all experiments in which the subjects were conscious of heat or cold, excessive moisture, or dryness, have been excluded from Table 3, with but two exceptions. In the two special cases the ventilation rate was under 2 cfm and therefore, the humidity inside the box was necessarily high. The box temperature was kept at a comfortable degree by cooling the psychrometric

room, rather than by excessive cooling of the ventilating current. Such a condition of high humidity would exist in any occupied room when the ventilation rate is reduced to 2 cfm per person, or less, regardless of moisture content in the air supplied. The impression of air quality will therefore be affected by the coexisting factors of excessive humidity and odoriferous matter in the air. To some extent this holds true with air flows up to about 5 cfm, unless the moisture content of the ventilating current is kept low, so as to maintain normal humidities in the occupied space, as was done in the experiments shown in Table 3.

Keeping this in mind, it can be seen in Table 3 that with ventilation rates under 5 cfm the air quality was poor to bad; at 10 cfm it was found to be good or fair, and at about 20 cfm it was excellent. Ventilation rates in excess of 20 cfm impaired the quality of air under the experimental conditions, owing

TABLE 3. SUBJECTIVE IMPRESSIONS OF AIR QUALITY IN RELATION TO AIR SUPPLY

(*Under comfortable conditions of temperature and humidity*)

OUTDOOR AIR SUPPLY CFM PER PERSON	NUMBER OF SUBJECTS RECORDING					CONSENSUS
	Excellent	Good	Fair	Poor	Bad	
1–2	..	..	..	..	2	Bad
4–5	..	1	1	4	..	Poor
6–8	..	2	2	..	..	Fair to good
9–11	2	3	5	1	..	Fair to good
12–16	1	3	1	..	..	Good
17–20	5	2	..	..	..	Excellent
21–35	..	4	1	..	..	Good
36–50	1	1	9	..	..	Fair

to an indescribable irritating sensation produced by non-chilling drafts. The objection was more serious with chilling drafts, but such experiments are not included in Table 3. This is clearly a limitation of the experimental method; the box was too small for the high air flows. Nevertheless, similar complaints often occur in ordinary rooms with ordinary distribution systems, when the circulation rate exceeds 20 cfm.

The judges' impression of air quality depended largely on the intensity of odor which they could smell to the fullest extent. The less the odor, the better the air. But since there was always a certain amount of odor in the box exhaust under all air flows by comparison with fresh outdoor air, the air of the box was never perfect in the opinion of the judges. This is, of course, a theoretical standard not easily realized in practice.

The outstanding point on which both judges and subjects agreed is that the quality of air is impaired when the outdoor air supply falls below 10 cfm per person even under the most favorable conditions of temperature and humidity. This is well in accord with the experience of the New York State Commission on Ventilation [3] and of Eberle and Raiss.[4]

[3] Loc. Cit. See Note 1.
[4] Eberle, C., and Raiss, W., Die Heizung von Schulgebauen. Beiheft zum Gesundheits-Ingenieur, 1931, No. 29.

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SUMMARY

A definite relationship was found to exist between intensity of body odor in the air of an occupied space and quantity of fresh air supplied, by confining human subjects in an airtight box and smelling the air coming out of the box.

With ventilation rates of 5 cfm per person or less, the odor was as a rule disagreeable and the air was regarded with disfavor by both judges and subjects, although the latter were not conscious of odor except for a sensation of staleness, flatness, or lack of freshness in the air as perceived by the sense of smell.

Under ordinary room temperatures (65-72 F), the minimum outdoor air supply per person necessary to prevent the accumulation of objectionable body odor in the air of the box varied from 13 cfm for subjects with body surface areas between 16.5 and 18.9 sq ft, to 27 cfm for surface areas between 19.0 and 20.5 sq ft. The average adult has a surface area of about 19.5 sq ft. Humidity had little or no effect.

Complete elimination of odoriferous matter in the air of the box could not be obtained with ventilation rates up to 50 cfm per person.

In the light of this preliminary study it would seem that the intensity of body odor in occupied rooms can be taken as a satisfactory index of minimum ventilation requirements. Practical requirements are now being studied in ordinary rooms using school children as well as adults for subjects.

DISCUSSION

D. D. KIMBALL (WRITTEN): Referring to the ventilation box experiments on odors, I am still doubtful of the efficacy of the measurement of ventilation by means of odors. Trained judges are not usually available, the reactions of different persons to odors will vary largely, and access to really pure outside air is not always directly convenient.

The experiments to follow in the schoolroom may be more enlightening. Could not the "irritating sensation produced by non-chilling drafts" be prevented by a different method of diffusing the air in the box? Much greater quantities of air per person per minute than here mentioned can be circulated through auditoria without noticeable drafts by proper means of diffusion.

Was any attempt made to determine the effect on the odor in the air leaving the box caused by thoroughly mixing with this air some fresh outside air? This should give some line on the effect of dilution and oxidization brought about by the admixture of the fresh air. How about ionization?

DR. E. V. HILL: This is rather an interesting presentation. It is the first I have heard of it. I have never been conscious of the fact that there was any connection between odors and air quantity. I have never been convinced that there is an odor, objectionable at least, in a normal, clean, human body. It seems to me that we ought to have some further information on the basis on which these tests were made. What kind of odors are these? Are they what are sometimes referred to as normal body odors? Is there such a thing as a normal body odor? The connection between air flow and odors is very vague, and I think very questionable.

L. A. HARDING: I believe that what Dr. Hill says is perfectly correct. I think garlic eaters might give a somewhat different index. This study has apparently corroborated former experience that approximately 10 cfm per person is the mini-

mum of outside air supply as is recommended by the A. S. H. V. E. Ventilation Standards.

DR. HILL: I think Mr. Harding brings the point out very clearly. Is this a test to substantiate somebody's previous view that 10 cu ft was essential, or is it based on a scientific investigation? If it is the latter, let's find some connection between body odors and air quantity. The man who has a coated tongue and hasn't had a bath might give off a rather objectionable odor. Clean, healthy individuals might give no odors whatever.

THOMAS CHESTER: It seems to me there is a very definite connection between the amount of fresh air required and the amount of odor given off per person. The amount emanated per person varies with activity. It is one thing in an office where people are sedentary, but it is much more pronounced in a gymnasium. In an occupied gymnasium there is a very pronounced odor of butyric acid. Dr. Hill says that we have to show it to be offensive, but it certainly cannot be said to be pleasant. Before the era of Pullman car ventilation anyone walking through a train had the edge taken off his breakfast appetite due to the odors encountered en route to the diner, and an individual with normal nostrils could detect a different odor in each sleeping car.

In this connection I remember very clearly that I was impressed by some of Dr. Hill's earlier work on school room ventilation, in which he stated that in order to properly rate odor intensity the olfactory nerves should be normalized by exposure to the outside atmosphere. For instance, you can not take a man in a box and get anything very definite by varying the air flow from 10 to 15 or 20 cfm, because he is more or less habituated to any prevailing smell and will not perceive any slight change. I remember that Dr. Hill said you have to go outside in order to get the nostrils normalized and then go into the room under investigation again, and in this way be able to get reliable perception of the odor condition.

DR. HILL: That is perfectly true in a classroom or in a large group where people of all kinds, races and ages are assembled. But my impression is that a normal, clean individual does not give off any odor. I would like to ask Professor Winslow's opinion on that.

C.-E. A. WINSLOW: I can only answer that by referring to the experiments at Harvard. This is the only study of the kind that has been made. As I understand it, these subjects were in the box and somebody outside noticed the odor; so that point was covered.

The results have been rather surprising to me, but I have regarded them as most interesting and significant. I wonder if there was one other problem that might present itself in applying these practical conditions. Here, as I understand it, you have almost solely the dilution value of this 10 or 20 or 30 cu ft that passes through the box. Would it not be true that in a room where there was a considerable air space you would get another factor in the way of the dissipation and the gradual disappearance of those odors that would give you somewhat less odor with the same air change than you would get in a case of this kind?

W. L. FLEISHER: I have not read the paper carefully, but my impression was that there were two rooms in which approximately the same number of people were enclosed, and that the same quantity of air, with absolutely the same entering conditions, or the same maintained conditions inside was produced. Then the change in the amount of recirculated air, and the amount of fresh air, was adjusted in one room, while the other room was being maintained definitely with all outdoor air; and then when the odor intensities in both rooms were the same it was decided that conditions were correct.

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I do not believe that has been brought out very carefully in this paper. I think that is a much more interesting experiment and much more to be relied upon than just the observation of the one room in which a certain quantity of fresh air was introduced.

H. L. ALT: I am a great believer in practicability. We, as ventilation engineers, have to install systems without knowing anything about the type of audience for which we are preparing our conditions.

If we could experiment with an average cross section of society, such as we find in schools and public places, we would have more practical conditions for our tests.

Suppose we grant that the clean man has no odor. If we take the average person and make a test of odor, it seems to me that the results will be of greater value.

W. H. DRISCOLL: Mr. Alt has brought up the point that I wanted to raise. We rarely provide ventilating equipment for individual persons. It is generally provided for assorted groups, and not for selected classes. There is no doubt in my mind that there is a definitely offensive body odor that comes from any person, even a normally healthy person.

Mr. Jones and myself subjected ourselves to a test in the box referred to in this paper. The test was made one morning shortly after I had arrived in Boston, after riding all night from New York on a smoky New Haven train. I hardly had a chance to wash decently, let alone take a bath. After I had been in the box a short time, Mr. Jones took a few whiffs and had some rather caustic comments to make about my condition. Now I may be pretty bad, but my health is good and my bodily functions are certainly not below normal. Nevertheless, the necessity for ventilation in my case was quite evident, and it is to meet such conditions that ventilation is required.

It seems to me that this paper is a particularly valuable contribution to the TRANSACTIONS of this Society. It is unfortunate, but readily understandable, that we do not appreciate the real significance, nor understand the real value of many papers that come before us. They pass through here so quickly that we haven't time, in the hurried presentation of the paper, to really get a broad understanding of what has been presented to us. We get a better appreciation of the real value of these papers later on when we have had an opportunity to sit down and analyze them and get from them the information that we can apply in the solution of the problems with which we are confronted.

It is very interesting to note that the results, to some extent at least, confirm the Standards of the Society as already adopted. I realize that a report of this kind, prepared from laboratory experiments, may lack certain elements that deprive it of conclusive practical value, but the fact of the matter is that it develops a technique that may be applied in making field observations and is, I think, a very admirable and very necessary step in that direction. I think the authors of this paper are entitled to the commendation of the Society for having taken that step, even though it may be considered as only a rather short step. It is a very significant and valuable paper, and I think the Society will come to realize that more and more as time goes on.

MR. FLEISHER: In actual practice there was a test of this kind made on a large auditorium in 1928, and it was very interesting as it confirmed in a way the observations that the investigators made. There was a theater in which an attempt was made to cut down the amount of outdoor air to the point where odors began to become objectionable. As I remember it, 6¾ cu ft of air per occupant became noticeably objectionable to anybody entering the theater, and the disagreeable odors started to disappear at about 9¼ cu ft of fresh air per person.

As far as I know that was the only practical observation or test made on an odor as a measure of the proper amount of fresh air. But that test was actually made in a theater seating about 3000 people, and the test was made over a considerable period of time. The minimum amount of fresh air that could be introduced without objectionable odors was 6¼ cu ft of outdoor air, and the threshold air quantity was somewhere around 9½ cu ft.

The interesting thing is that this type of test can be made on hundreds of theaters because it is easy to regulate the amount of fresh air to recirculated air in these theaters. It might be of interest to have the Research Laboratory or some special committee make actual tests both during the summer and winter to find out just what the proper quantity is for the removal of odors.

That is not a difficult thing to do as a field study. It is one of the few cases where we have a very easy chance of making a field study to check the observations in the laboratory.

MR. ALT: I have always understood that when we recirculate air we have to supply 25 per cent new air to avoid objectionable odor, especially in theater work where the total amount circulated is 20 to 30 cfm. It is very seldom that a theater circulates less than that, and it is unusual to circulate more than 30 cfm. This means 5 to 7½ cu ft of fresh air per occupant per minute. I have always understood, and it has been borne out in practice, that 25 per cent is the very minimum amount of fresh air that can be supplied without having objectionable odors. If there is more than that, it is safer. That bears out Mr. Fleisher's experiment.

MR. FLEISHER: There is a new factor arising in that particular thing which is of importance. Today there are two methods of cooling: one with surface coolers and one with washers, and I think very definitely you will find that the amount of outdoor air required for dilution will vary considerably and will be lower with the washer than it will be with the surface cooler; that is, where both fresh air and recirculated air are passed through the cooler. So you would have to bear that in mind very carefully in making observations or forming an opinion. The washer does something in the way of removing odors which the surface cooler does not.

G. R. WAIT[5]: While investigating particles from the breath, which I did by counting the particles that acted as centers for condensation of water vapor, I found rather definite evidence that the body, as well as the breath, gives off very minute particles of matter. I used a small closet in one experiment; I kept my nostrils outside and took in and expelled air outside the closet. At the same time I placed my body inside the closet. I found that the numbers of particles in the closet increased in number and the only way I could account for this fact was to assume that the particles were given off by the body. I did not extend these observations so that I do not yet know a great deal about the particles other than the mere fact that they are given off by the body.

I wonder if it is possible that the number of particles given off has any relationship to body odors? If so, then the counting of particles would provide a method of measuring the intensity of odors instead of having to depend upon the sense of smell. If investigations should show that the intensity of an odor is controlled by the number of particles given off by an odoriferous body, then the counting of particles could be used as a basis of classifying the intensity of odors, instead of the present method, which at best is subject to considerable personal bias. Certainly this matter deserves to be studied by those interested in the perception and classification of odors.

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[5] Dept. of Terrestrial Magnetism, Carnegie Institution of Washington.

MR. CHESTER: Our nostrils are not as good as those of a savage, but I believe anybody who says there is no odor from the human body is wrong. There is certainly some emanation from the body in general and especially from the feet. A bloodhound can track a man after a lapse of some hours, and I claim that that shows very definitely there is an emanation from the human body which can be detected by the sense of smell. To a dog each human being has a different odor, easily perceptible to his nostrils. Although our olfactory nerves are less keen than those of a dog, they are capable of detecting body odors, unless they are blunted or impaired by a cold in the head or unless they are atrophied entirely. The soap manufacturers do not agree with Dr. Hill, judging by the vast expenditures they make in B.O. or body odor advertising.

DR. HILL: This is supposed to be a scientific body, and my thought was, when I criticised the paper perhaps too harshly, that it was a rather lax method, to put a clothed subject into a closed box and try to find out how much air was required to reduce odors by the sense of smell.

We have been doing experimental work for the past four years. There is a cabinet in our laboratory which contains 44 cu ft of space. We have had 14 female subjects, most of them girls from the Art Institute. We have repeatedly placed these girls in the cabinet for as long as 1½ hours with recirculated air, without a particle of outside air being introduced, and at the end of that time could detect no objectionable odor whatever. Out of the 14 subjects there was one of them that gave a definite odor, and it was interesting because the other girls in the test series objected to continuing the work with this girl. They said she wasn't clean, and so we let her go.

But, I am firmly convinced in my own mind that a nude subject, healthy, and clean, gives off no perceptible odor. As to subjects in a cabinet with clothing on, the clothing may be partially saturated with moisture and may give off odors, and you may have conditions about the body which are abnormal which give off odors.

I remember very distinctly being called into Mayor Thompson's office a good many years ago to determine the defects of the ventilation system in his office. It was very well ventilated as far as air change was concerned, but he complained of an objectionable odor. After a long series of tests of one kind or another, we finally decided that a dye in the rugs was the cause of the trouble.

All of these things must be taken into consideration. If you are discussing the question of odors of an audience, it is a very logical thing to find out how much air is required to remove the odors that arise in a mixed audience of this kind. But, when you are talking about one individual, I think that clothing and physical condition of the individual is the determining factor and not the question of odor itself.

MR. HARDING: In regard to a point that was raised by Mr. Alt regarding the circulation of 20 cu ft per person per minute, the amount of air circulated depends entirely on the allowable temperature drop between room outlets and the room temperature, and the sensible heat to be removed from the room. An 8 F drop and no sensible heat emitted in the room except that which is given off by the occupants, requires the circulation of approximately 26 cfm to remove this heat.

In addition to the sensible heat furnished by the occupants, other amounts of sensible heat must be removed and in some cases you have to circulate considerably more than 30 cfm per occupant.

MR. DRISCOLL: I always knew that Dr. Hill had a great nose for news, but I am beginning to suspect that he has lost his sense of smell.

JOHN JAMES: A part of this discussion has centered around the principle or method of measuring the intensity of odors rather than depending upon the human sense of

smell. Mr. Wait has indicated the possibility of counting the number of particles emanating from the breath and relating the quantity to some definite classification of body odors.

It is interesting to note that other investigators are developing procedures whereby an odor under observation is carried to the olfactory membrane by direct injection into one or both nasal passages. The apparatus consists of a group of test bottles, glass syringes and suitable nose pieces for the direct introduction of the odors. A regulating or exit valve is arranged so as to release the odor from the test bottle upon application of a slight finger pressure of the individual performing the test. By this method the volume of injected air plus odor can be measured and the injection may be made under known pressure as well as in an optimum direction in order to reach the more sensitive olfactory membrane.

Investigation of odors in this manner has been made on several people and with many odorous substances. The smallest volume of odor delivery to the nasal passage necessary for identification of an odor has been termed the olfactory coefficient. The results of all of these tests indicate for normal individuals that the olfactory coefficients for the several odors tested were about the same.

It is conceivable that this method might be adapted in some way to the studies referred to in this paper. Samples of room air at various concentrations could be collected in test bottles and injected directly to the nasal passage of an individual who would then be able to classify the odor according to some previously determined scale. It is appreciated that this procedure when adapted to body odors would require more accurate odor perception than the method described for determining the olfactory coefficients of substances. However it is probable that a comprehensive study might develop a more accurate means of determining odor intensity than has thus far been established.

W. H. LEHMBERG: With reference to Dr. Hill's first question, we had 10 male subjects, all normal individuals, Harvard students, and as far as possible we controlled personal factors which included baths, changing of underwear, etc. None of our subjects ate onions or garlic before being subjected to the tests, so I think we can say that we had average individuals with average personal habits of hygiene and eating. In regard to the question raised about the practicability of advancing it as a code, we do not propose to do that yet because at the present time we are working on this problem with adults, adolescents and children, in regular size rooms, and we hope that when we get the data together we will have something more tangible. After establishing the requirements of temperature and humidity, odor intensity will no doubt be the next criterion of good ventilation.

The curves given in this paper represent the impressions of the judges. The base or zero odor was that of the air in the conditioned room in which the experimental box was placed. The psychrometric room was ventilated with outdoor air at the rate of 750 cfm per person. I think that that is a reliable base for an approximate zero odor intensity. Comparisons were made by not less than three judges in the outlet pipe of the box which ran back into the exhaust duct of the air conditioning system and discharged out of doors.

Fifteen judges, as many as we could get, estimated the strength of the odor and their consensus of opinion was plotted against the air flow for each test.

DR. HILL: May I ask one more question? Did the judges of the odor know that this air was coming from around the body of the subject?

MR. LEHMBERG: Yes, they did. The judges knew that the air was coming from the subject because this box which sat inside the psychrometric room had one duct going into it at the head end and dispersed through the box by means of a perforated manifold. Although the judges did know that the box was ventilated, they did not know the amount of air flow nor the test conditions.

XUM

MR. ALT: The quantity of air depends upon the amount of heat to be removed and the temperature drop in the air when supplied. We always use 15 F in theaters that I have worked on, and I believe that throughout the country you will find more supplying 20 than 30 cfm.

MR. LEHMBERG: The doubts expressed by Mr. Kimball substantially coincide with those entertained by us when this work was started. Moreover, our major purpose was to answer this question. I believe that all points pertaining to it are answered satisfactorily in the text.

Mr. Kimball's second question will probably become self-explanatory by correcting a typographical error which appeared in the reprints available at the meeting, so as to read, "in excess of 20 cfm" instead of "in excess of 10 cfm."

Questions three and four were outside the scope of the present study.

No. 1010

THE HEAT REQUIREMENTS OF BUILDINGS

By J. H. WALKER * AND G. H. TUTTLE ** (*MEMBERS*)
DETROIT, MICHIGAN

THIS paper is a contribution to the work of the Technical Advisory Committee on Heat Losses from Buildings. Its purpose is to make available to Society members certain data, relative to building heat losses as reflected in the actual consumption of steam, which have been compiled by district heating companies.

The primary objective of the above-mentioned committee is to study and improve the methods of calculating heat losses for the design of heating systems. The question of the actual heat consumption of buildings is only incidentally related to that objective but is of much practical importance. Its value is in the prediction of heat consumption, rather than in the design of heating equipment, and it is of economic rather than technical significance. The figures which will be given have been obtained under practical conditions and are therefore influenced by uncontrollable factors, such as human habits, and by unmeasurable conditions, such as local vagaries of the weather. Therefore, one should not expect to find extreme refinement or consistency in the data.

STEAM CONSUMPTION AND THE DEGREE-DAY

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically, it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association* determined from experiments in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F.

As a result of the information initially provided by the gas industry and later qualified and used in other methods of heating, a unit known as the degree-day was originated and is now used almost universally as a means of estimating and comparing fuel consumptions for different outdoor tempera-

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tures. It is defined as a difference of 1 deg between the base inside temperature and the mean [1] outdoor temperature for a day. The number of degree-days in any period is the summation of the differences between the base temperature and the daily mean temperature for that period.

Establishing the Base Temperature

Recently the *National District Heating Association* has studied the metered steam consumption of 163 buildings in 22 different cities, and has published [2] data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct. In the *N. D. H. A.* study, the steam consumption of each building by months was divided by the number of days in each month, thus giving the average daily steam consumption by months. The average steam consumption was then plotted against the average monthly

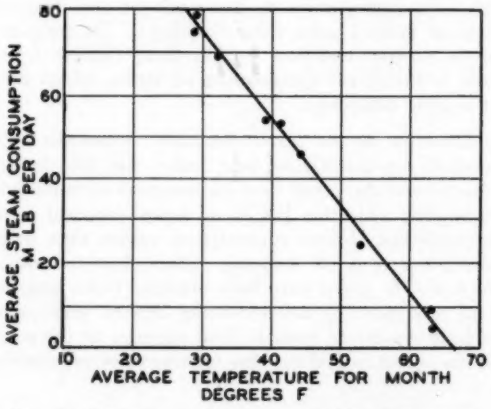

FIG. 1. METHOD OF DETERMINING BASE TEMPERATURE FOR DEGREE-DAY CALCULATIONS

temperature, as shown in Fig. 1, and the temperature at which a line drawn through the points crossed the base line indicated the temperature corresponding to zero steam consumption, or the base temperature. The composite results from 163 buildings calculated in this manner are shown in Table 1.

The grand average of 66.0 F is close to the *A. G. A.* figure of 65 F. It will be noted that the base temperature calculated for hotels, apartments and residences is consistently higher than those for such buildings as garages, auto sales buildings, and manufacturing buildings. This, of course, would be expected in view of the higher inside temperatures carried in the former group; in fact, an even greater difference would be expected. For an all-round average figure, the *A. G. A.* base of 65 F may therefore be safely used, and if greater refinement is desired, the figure for the type of building in question can be taken from Table 1.

[1] Average of the daily maximum and minimum temperatures.
[2] Report of Commercial Relations Committee, 1932 Proceedings, *National District Heating Association.*

A practical application of the use of 65 F base for comparing steam consumptions for heating is shown in Fig. 2. In this illustration the steam sold by a district heating company per month is plotted against the total degree-days for the month. It will be noted that the relationship is fairly constant for all temperatures but that the curve does not intersect zero on the abscissa. However, if a base of 66 F were used, as is established in Table 1, the curve would approach zero on both coordinates.

TABLE 1. BASE TEMPERATURE FOR THE DEGREE-DAY

TYPE OF BUILDING	NO. OF BLDGS. ANALYZED	TEMP. CORRES. TO ZERO STEAM CONSUMPTION
Office	60	66.2
Office and Bank	4	65.8
Bank	3	66.2
Office and Telephone Exchange	2	65.5
Office and Stores	6	67.4
Stores	11	64.0
Department Stores	12	64.3
Hotels	7	66.5
Apartments	14	68.8
Residences	8	66.9
Clubs	4	65.5
Lodges	5	64.9
Theatres	3	67.6
Churches	2	65.8
Garages	2	64.8
Auto Sales and Service	4	61.2
Newspaper and Printing	3	67.7
Warehouse and Loft	3	67.7
Office and Loft	2	65.2
Manufacturing	8	65.4
Grand Average for 163 Buildings	..	66.0

Mean and Average Temperatures

Since the U. S. Weather Bureau publishes mean daily temperatures rather than average temperatures for the 24 hours, the temperature used in defining the degree-day is the *mean* temperature. Daily variation between mean and average temperatures may be considerable, but when considered over a period of years the difference is quite small. A summation of the total degree-days per month and per year, calculated in each manner, is shown in Table 2.

The total number of degree-days per year is approximately 0.6 per cent lower when calculated on the mean temperature basis than when calculated on the average temperature basis, indicating that it makes very little difference which figure is used.

STEAM CONSUMPTION OF BUILDINGS

Table 3 gives the steam consumption per degree-day, expressed in three different ways, for 196 buildings in 14 different classifications. These buildings

are divided among 21 different cities in the United States. The steam used for heating the domestic water is included in these figures, but in the case of office buildings, the steam for heating only is also shown. The data are placed on a comparable basis by expressing the steam consumption in terms of pounds per degree-day per thousand square feet of equivalent installed radiator surface, per thousand cubic feet of heated space, and per thousand Btu of calculated heat loss.

The choice of these units of comparison needs some explanation. The use of *heated* space in preference to the gross cubage used by architects is obviously more accurate for this purpose. The architect's cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. At best, the

TABLE 2. DEGREE-DAYS CALCULATED ON A 65 F BASE, USING MEAN DAILY TEMPERATURES AND AVERAGE DAILY TEMPERATURES

MONTH	DEGREE-DAYS BASED[a] ON MEAN TEMP.	DEGREE-DAYS BASED[a] ON AVER. TEMP.	PER CENT OF TOTAL DEGREE-DAYS PER MONTH, BASED ON MEAN TEMP.
January	1199	1189	19.16
February	1008	1043	16.10
March	943	961	15.07
April	576	572	9.21
May	240	226	3.83
June	7	3	0.12
September	55	57	0.88
October	375	384	5.99
November	748	749	11.96
December	1106	1113	17.68
Total	6257	6297	100.00

[a] U. S. Weather Bureau temperature records for Detroit, 1924-1934.

cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents.

The use of radiator surface as the basis of comparison has two objections. One is that the amount of radiator surface in a building is often either excessive or deficient, and figures for steam consumption based on it are therefore likely to be in error. Another reason is that it is difficult to convert fan coil surface into equivalent direct radiator surface with accuracy. On the whole the use of radiator surface as the basis of comparison is the least satisfactory of the three methods.

The authors believe that the calculated heat loss affords the most satisfactory basis of comparison. The figures in the last column of Table 3 were obtained by dividing the steam consumption for the heating season by the degree-days and by the Btu heat loss, in thousands, as calculated for a 70 F difference between inside and outside. Unfortunately, the heat loss figures are not always available and were not available for all classes of buildings given in the table.

It should be noted that the figures in Table 3 are for the heating season only, and include steam for heating domestic water. To calculate the addi-

tional steam used only for heating water, in the summer months, the data on hot water given later in this paper can be used.

This method of estimating does not afford absolute accuracy. It gives approximate figures varying perhaps plus or minus 10 to 15 per cent from the true figures.

Monthly Variation in Steam Consumption per Degree-Day

The data on consumption per degree-day given in Table 3 are calculated for the entire heating season rather than for the individual months. The figure as

TABLE 3. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS[b] (HEATING SEASON ONLY)

BUILDING CLASSIFICATION	NO. OF BLDGS. LISTED	STEAM CONSUMPTION, POUNDS PER DEGREE-DAY — 65 F BASIS[e]		
		Per M Cubic Feet of Heated Space	Per M Square Feet of Radiator[d] Surface	Per M Btu per Hour of Heat Loss[e]
Apartments	16	1.78	97.5	0.359
Hotels	10	1.46	80.6	0.371
Residences	12	1.32	64.2	.....
Printing	7	1.26	105.5	.....
Clubs and Lodges	10	0.96	77.0	.....
Retail Stores	18	0.90	80.6	0.268
Theatres	6	0.90	75.0	0.498
Loft and Manufacturing	16	0.89	72.3	0.283
Banks	7	0.88	45.2	.....
Auto Sales and Service	8	0.83	62.2	.....
Churches	6	0.58	49.4	.....
Department Stores	14	0.57	60.7	0.238
Garages (Storage)[f]	6	0.42	72.3	.....
Offices (Total)	35	1.09	70.0	0.283
Offices (Heating only)	35	0.975	65.4	0.256

[b] Includes steam for heating domestic water for heating season only.
[c] Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).
[d] Equivalent radiator surface.
[e] The figures are a numerical—not a weighted—average for the several buildings in each class.
[f] Based on zero consumption at 55 F.

given, then, is the average consumption figure for the entire year. The consumption factor per degree-day is not always constant throughout the heating season. In St. Louis, Mo., for example, the steam consumptions of 140 buildings were studied for three consecutive years. The monthly consumption factors per degree-day per thousand cubic feet of heated space are shown in Fig. 3.[3] It would appear from this study that the steam consumption per degree-day for heating only is lower in the spring and fall months than during the winter months. However since September and May represent less than 5 per cent of the entire heating season, the error would be slight if the same consumption factor were used for these months as is used for the remainder of the year.

[3] R. M. McQuitty, as reported in the *N. D. H. A. Proceedings* of 1934, by the Commercial Relations Committee.

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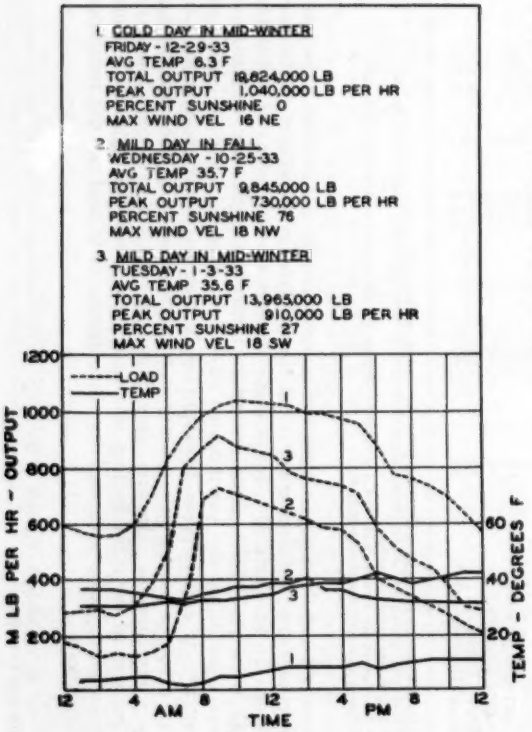

FIG. 4. TYPICAL LOAD CURVES FOR A DISTRICT HEATING SYSTEM

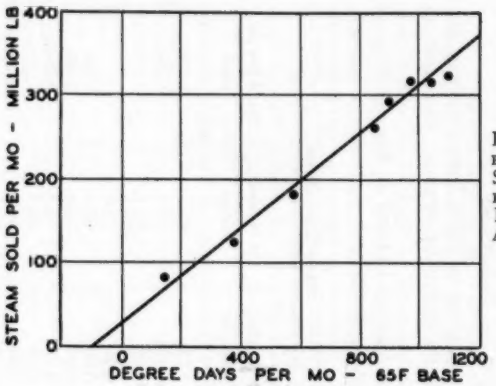

FIG. 2. RELATION BETWEEN STEAM SOLD PER MONTH BY A DISTRICT HEATING COMPANY AND DEGREE-DAYS BASED ON 65 F

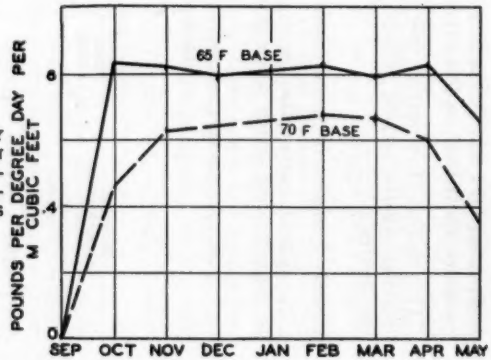

FIG. 3. MONTHLY VARIATION IN STEAM CONSUMPTION PER DEGREE-DAY FOR ST. LOUIS BUILDINGS

Effect of Temperature Control Devices Upon Steam Consumption

When temperature control equipment is installed in existing buildings, it is usually for the purpose of reducing the steam consumption. Table 4 gives the steam consumption in pounds per degree-day for 51 buildings before and after installing temperature control equipment. The reduction in steam consumption expressed as a per cent saving is given in the last column.

TABLE 4. COMPARISON OF STEAM CONSUMPTION BEFORE AND AFTER INSTALLING TEMPERATURE CONTROL EQUIPMENT

TYPE OF CONTROL	NO. OF BLDGS.	STEAM FOR HEATING, POUNDS PER DEGREE-DAY		SAVING, PER CENT
		Before	After	
1. Sub-atmospheric Pressure System	5	1720	1450	15.7
2. Orifice System Controlled by an Outdoor Thermostat	2	2039	1600	21.6
3. Orifice System Controlled by Group of Outdoor Thermostats	1	4580	4255	12.2
4. Orifice System—Manual Control	3	7930	6750	14.9
5. Orifice System—Electric Zone Control	2	4110	4025	2.1
6. Intermittent Control—Electric Timer	5	1340	1200	10.4
7. Intermittent Control Actuated by Outdoor and Radiator Temperatures	14	772	623	19.3
8. Central Thermostat	19	776	585	24.6

TABLE 5. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS

BUILDING CLASSIFICATION	LOAD FACTOR	POUNDS OF DEMAND PER HOUR PER SQUARE FOOT OF EQUIVALENT INSTALLED RADIATOR SURFACE
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theatres	0.126	0.151

The results in Table 4 should be interpreted with caution. The data by no means constitute a comparative test of control equipment, because in many cases the reduction in steam consumption following the installation of control equipment cannot be credited entirely to the new equipment. Very large percentages of saving are always open to question as it is likely that in most cases the control equipment is only one of the several factors which have com-

bined to produce the saving. The results, however, indicate in a general way what can be accomplished with control equipment coupled with intelligent handling of a heating system.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 5.

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 5, the theatres, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

The maximum steam demand for heating of a building does not necessarily occur on a day of minimum temperature. There are apparently other factors, such as direction and velocity of wind, sunshine, and time of year, which affect the steam consumption of buildings. This peculiarity of peak demands is illustrated in Figs. 4 and 5. In Fig. 4 are three typical daily load curves for a district heating system which supplies steam to approximately 1300 buildings. They show the steam delivered to the street mains by the boiler plants. Curve 1 is the system load for a cold day in mid-winter; Curve 2 is a mild day in the fall; and Curve 3 is for a mild day in mid-winter. The two days on which loads for Curves 2 and 3 were established were practically identical as far as apparent heating requirements were concerned. It is true that there was a larger percentage of sunshine on the October day; however, the peak output which occurs between 8:00 and 9:00 a. m. is very little affected by the sunshine during the day. The peak output on the January day was 25 per cent higher, and the total output for the day was 42 per cent higher, in spite of the fact that the 6:00 a. m. temperatures were only 1 deg different and the average temperatures for the two days were the same. The explanation probably is that those in charge of the operation of some building heating systems are inclined to delay in turning on heat in the early fall days. It is also probable that people can tolerate a cool day in the fall without heat more readily than they can in mid-winter.

Curve 1 is for a cold day in mid-winter. On this particular day, when the average temperature was approximately 30 deg lower than the mild day in January, the peak output was only 14 per cent higher and the total output for the day 42 per cent higher. It is evident from a study of these curves that outside temperature is by no means the sole factor in determining heating requirements, particularly the maximum demand.

The points plotted in Fig. 5 are maximum steam output figures of a district heating system with given 5:00 a. m. temperatures. The days of the week are denoted on the curve by code. One might assume that office buildings which are unheated on Sundays would require more heat on Monday than on any other day of the week having a corresponding temperature. If this were the case, there would be a predominance of Monday points at the top of the group. This, however, is not the case. Here again the data prove the fallacy in the belief that there is a definite relationship between peak steam demands and outdoor temperatures. It will be noted, however, that the relation between peak demand and outdoor temperature is much more constant in the lower than in the upper ranges of temperature. The curve which is located so as to

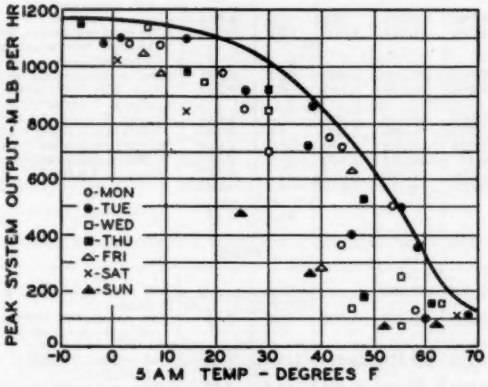

FIG. 5. RELATION BETWEEN PEAK SYSTEM OUTPUT OF A DISTRICT HEATING SYSTEM AND THE 5:00 A. M. TEMPERATURE. ALL DATA TAKEN BETWEEN JANUARY 1ST AND JUNE 1ST

include all the points is used to predict the boiler capacity which must be held available for the day's load.

A careful study has been made to correlate the several variables which might control these daily peaks in the hope of determining a consistent relationship which would make it possible to predict the boiler requirements. Temperature, sunshine, wind velocity, day of the week and season of the year have all been considered, but attempts at correlating them have been unsuccessful. There are probably certain psychological or physiological factors which are influential and which cannot as yet be identified.

HOT WATER REQUIREMENTS OF VARIOUS CLASSES OF BUILDINGS

There is very little published information on the actual amount of hot water used by various types of buildings. In order to determine the proper size of water-heating economizer for a building, it is necessary to know the amount of hot water used and also the rate at which it is used at different times of the day. These figures are also needed for determining water heater sizes

TABLE 6. HOT WATER USED BY VARIOUS TYPES OF BUILDINGS

CITY	BLDG. VOLUME[5] CU FT	HOT WATER USED, LB PER DAY	MAXIMUM HOT WATER DEMAND, LB PER HOUR	LB HOT WATER PER DAY PER M CU FT BLDG. VOLUME[5]	MAXIMUM HOT WATER DEMAND LB PER HOUR PER M CU FT BLDG. VOLUME[5]	NOTES
Office Buildings						
Baltimore..	5,500,000	33,600	3,375	6.1	0.613	Utility offices; 3 beauty shops, 6 doctors' offices; 3 shower baths; hot water in every room.
Baltimore...	2,790,000	26,800	2,500	9.6	0.896	24-hour restaurant open 6 days per week, 300 meals per day; 1 small soda fountain and lunch counter.
Baltimore...	795,000	2,100	750	2.6	0.944	Bank on 1st floor.
Baltimore...	4,361,286	46,000	4,310	10.5	0.989	1–24 hour restaurant, 400 meals per day; 1–16-hour restaurant, 200 meals per day; 1 barber shop.
Detroit.....	3,480,000	38,740	3,360	11.1	0.965	Small restaurant on 1st floor; 1 barber shop.
Detroit.....	1,776,000	29,035	1,690	16.4	0.952	Drug store with soda fountain; 3 beauty shops; 1 barber shop.
St. Louis...	4,336,095	67,910	7,000	15.7	1.616	116,000 lb hot water used on maximum day.
St. Louis...	1,617,079	7,880	1,200	4.9	0.743	12,000 lb hot water used on maximum day.
St. Louis...	2,417,385	32,700	4,920	13.5	2.040	45,000 lb hot water used on maximum day.
St. Louis...	3,196,497	30,265	2,800	9.6	0.878	56,000 lb hot water used on maximum day.
Portland....	531,000	14,500	1,265	27.3	2.380	Stores on 1st floor.
Portland....	1,462,000	13,770	1,015	9.4	0.694	
Average..				11.4	1.14	
Hotels						
Akron.....	7,000,000	292,200	....	41.8	....	450 rooms, kitchen, laundry, and barber shop.
Detroit.....	940,000	26,110	2,500	27.8	2.66	234 rooms, 80 baths, restaurant, barber shop, soda fountain, 106 guests.
St. Louis...	1,253,193	145,575	14,080	116.0	11.21	345 rooms, 198,000 lb hot water on maximum day.
Portland....	2,175,000	137,985	15,775	63.5	7.25	297 rooms, laundry, dining room, and Turkish bath.
Portland....	1,163,000	51,105	4,800	44.0	4.13	Dining room and coffee shop.
Portland....	212,000	12,235	1,100	57.7	5.19	Apartment hotel.
Average..				58.5	6.09	
Apartment Building						
Portland....	635,000	40,210	3,605	63.3	5.68	
Women's Club Building						
Detroit.....	901,950	70,365	4,180	78.0	4.63	210 sleeping rooms, dining room, and laundry.
Restaurant						
Detroit.....	....	34,680	2,310	....	....	24-hour restaurant; seating capacity, 128; meals per day, 2,500.

[5] Heated Space.

and estimating steam demands caused by water heating. To obtain this information,[4] water meters were installed to meter the cold water supplied to water heaters in 20 buildings in 5 cities. The results are given in Table 6.

The average hot water requirement of the twelve office buildings was 11.4 lb

[4] 1932 *Proceedings of the N. D. H. A.*

per day per thousand cubic feet of building volume; however, in one building in Portland, Ore., it was as high as 27.3 lb per day per thousand cubic feet.

The average amount of hot water used by hotels was 58.5 lb per day per thousand cubic feet, but in one case the figure was as high as 116.0.

The last column of Table 6 gives the maximum demand for hot water in terms of building volume. These figures should be used in determining water heater sizes and expected steam demand. In using the figures given in the table to determine steam demands and consumption, it can be safely assumed that one pound of steam will supply 10 lb of hot water. To convert Table 6 to read pounds of steam instead of pounds of water, the water constant given may be divided by 10.

SUMMARY

The paper gives the heat consumption of a large number of buildings on a degree-day basis and recommends, as a unit for comparison, the heat consumption per degree-day per thousand Btu of calculated heat loss.

The maximum demands and load factors are given for different kinds of buildings. That the maximum demand and heat consumption are affected by other variables than temperature alone is demonstrated by typical load curves from a district heating system.

Data on the hot water consumption of various kinds of buildings are included.

DISCUSSION

P. D. CLOSE (WRITTEN): This paper contains much valuable and interesting data relative to steam consumption of various types of buildings per degree day. The paper points out that, "The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated." Because of the importance of wind velocity, it seems to be illogical to make fuel consumption comparisons on the basis of degree days which take into consideration the temperature head only, and neglect entirely the wind velocity which may in many cases be equally or more important than the temperature head. In other words, it is quite possible for the fuel consumption to be at least doubled as the result of an increase in the wind velocity, for the same temperature head or number of degree days.

In a certain specific case the heat loss of a room was doubled for the same outside temperature when the wind movement changed from 0 mph to about 13.3 mph, and tripled for the same outside temperature (or number of degree days) when the wind movement increased from 0 mph to 26.7 mph.

Although this was a calculated result based on THE A. S. H. V. E. GUIDE data, it is generally recognized that the heat loss of a building can be changed considerably as a result of an increase in the wind velocity without any change in the outside temperature, particularly in the case of loosely fitted windows. Under such conditions, it appears that the fuel consumption per degree day which neglects wind velocity, might be considerably in error.

F. B. ROWLEY (WRITTEN): This paper is a valuable contribution to the literature on the subject of heat requirements of buildings. It brings out some very interesting points in regard to the various methods of estimating heat losses and, in particular to the base temperatures used for calculating the degree days for the heating season.

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It is of particular interest to note the close agreement between the 66 F base temperature as arrived at by the authors and the 65 F base temperature used by the *American Gas Association* for determining the degree days. Upon first thought it would seem that 70 F would be the logical temperature to use for making this calculation, but since the facts show differently, there must be some reason for it.

Since this method of calculation takes into consideration only the temperature differences, it would seem to prove that the method of calculating heat losses by temperature difference alone gives excessive values. This might be charged against the coefficients of transmission used or against some factors which have been omitted. In my opinion, it is due to the latter. There has always been some question about using mean temperatures as recorded by the Weather Bureau instead of average temperatures as a basis for heating loads. Table 2 of the present paper shows conclusively that for seasonal conditions there is practically no difference.

One element left out of this method of calculation is that of sun effect on the building. It seems quite probable that the amount of heat absorbed by the average building from the sun would be sufficient to take care of the difference in heat loss when calculated on the 70 F basis as compared with that calculated on the 65 F basis. The effect of sunshine varies between wide limits, depending upon the day and the type of building, but for average conditions it must certainly reduce the heating load and its effect is worth considering as a possible reason for the apparent discrepancy between actual heat loads and calculated heat loads when using the 70 F inside temperature.

From the curve of Fig. 3 the total steam consumption per degree day based on 70 F temperature runs more uniformly throughout the season than when based upon 65 F temperature. This would indicate that the 70 F base was theoretically correct and that some other factor was affecting the losses. Incidentally, it would seem that the two curves should be reversed in this figure and that the pounds per degree day would be greater for the 65 F base than for the 70 F base. However this may be due to a misinterpreting of the figures.

From the last column in Table 3 it is interesting to note that such buildings as retail stores, department stores, office buildings (heating only), and loft and manufacturing buildings run very close in steam consumption based on calculated values. On the same basis, apartments and hotels run about 40 per cent higher, indicating that it is much more difficult to control the output from these buildings.

Referring to Fig. 4, the increase in steam consumption for the winter day, indicated by Curve 3, over that for the fall day, indicated by Curve 2, may be partly due to the temperatures of the previous day. Curve 3 indicates a rising temperature throughout the period, and Curve 2 a slightly falling temperature. If this condition had existed throughout the previous day, there would have been more heat stored in the structure to help take care of the heating on the fall day than would have been the case for the winter day. Likewise due to the lag in the heat capacity of the building, the higher temperatures for the end of the winter day as compared with those for the end of the fall day would not have had the full effect on the heating of the building, although they enter into the average.

The above remarks are not intended as criticism but rather to bring up some of the points which have occurred to me in reading the paper. The authors are to be congratulated in the analysis they have made and in the method of presentation.

D. S. BOYDEN: The study of the heat loss from buildings, which has been undertaken individually for a great many years and has never yielded a satisfactory conclusion, is now being studied by this Committee on Heat Requirements of Buildings and a large number of members in the field, with one objective in mind, that it may be possible to evaluate the various factors which will make a complete formula for correctly and accurately computing the heat loss from buildings.

We know that much work has been done in the transmission of heat losses through various kinds of material. No values are available based on the cubical contents of the building until we begin to add the other factors of infiltration, ventilation, sun effect and the many other intangibles, the value of which we are now trying to determine.

This paper, together with others, is gradually accumulating sufficient data to bring about the true value of these variables. It is apparent that the statistics compiled by the *National District Heating Association* can be of great value. We can assume any day or any kind of weather that this Committee determines is desirable to take, and compare it with the other days. Days of high wind velocity with equal temperature, as Mr. Close has suggested, may also be used.

Mr. Walker indicates a great variation in the amount of steam per degree-day in the spring and fall. That would be very true in homes, or individually heated buildings in which there is an isolated plant, and where the owner may not wish to start the heating plant. In those cases where the building is supplied with central heat simply by the turn of a valve by the janitor, this factor might not have such an influence.

In the fall we become acclimated to much lower indoor temperature with comfort. This condition has been particularly noted in district heating where a large number of retail stores occupying the first floor of a building are supplied with central steam.

However, after the season has become well advanced and heating is required continuously, the first chilly draft from open doors affects our comfort, and we continue to use steam extravagantly, until we have reached our newly acclimated condition.

L. A. HARDING: This paper tends to confirm the belief of most heating and ventilating engineers that the heat loss of the building is not a matter of very precise calculation. Miraculously, as it were, from 1932 to 1933, the estimated heat transmission loss of buildings increased overnight, on the appearance of THE 1933 GUIDE.

Many factors enter into the calculations for the heat loss from buildings that are difficult to evaluate. I recently found that the calculated heat loss of a large office on a zero day was about 7000 Btu per hour. The body heat delivered by the occupants plus the heat equivalent of the illumination was estimated as 11,000 Btu per hour.

ALBERT BUENGER: I was very much interested in the curve in Fig. 3 which the author presented, showing a difference in the 65 and 70 F base. A personal experience in my own home recently brought this difference to my attention conclusively.

In my own house I had been using fuel oil, and discovered that over a number of years the requirements were about one quart of oil per degree day. One fall an enterprising fuel oil dealer told me that he was selling oil on an automatic heating service. He was figuring the oil on a degree day basis.

I told him of my experience in using one quart of oil per degree day, and he said that he would supply the oil on that basis. After a few months he advised me that according to his calculations, I only needed 750 gal of oil instead of 2000 gal. He had changed his method of computation to a 70-deg base instead of a 65-deg base.

Experience showed that in the fall months when he was getting his principal data, the temperatures were too low, resulting in a degree base which was too low, so by moving it up to a 70 F base, a more uniform oil consumption for the entire year was obtained. The data this particular fuel oil dealer is obtaining will be valuable information as to oil consumption per degree day.

C. C. TRUMP: There are two questions I should like to ask about the data. One is whether any of the buildings were observed as to the color? In tests we made

in oil refineries, we found that black tanks reached a temperature in the sun of 140 F; aluminum painted tanks never went over 95 F. That is a difference of 45 deg. Also, was the moisture or wetness of the building walls measured? My understanding is that a new concrete building often takes 30-50 per cent more heat than one that has dried out.

MR. BOYDEN: The last speaker has just brought up a very important matter which is generally overlooked and which is assumed to be poor operating practice.

A new building, either concrete or steel, will use materially more steam. I think 50 per cent is a little high, 15 or 20 per cent is more nearly correct. This diminishes regularly for the first three years, when practically all the voids are filled which reduces infiltration.

In regard to the spring and fall degree day, we can very well neglect the first month at each end of the season, or the first one in the fall, and the last one in the winter, but still adhere to 65 F in commercial buildings. In residences where there is 100 per cent occupancy, we might very properly use 70 F, but when we get into a large number of buildings with certain spaces unoccupied and where people are not as particular about the temperature, the 65-degree day is sufficiently accurate.

J. H. WALKER: The relation of this paper to the general problem of heating requirements is not to be overlooked. The aim of the Committee on Heat Requirements of Buildings, as I understand it, is to improve methods for calculating the maximum heat requirement of a building and to provide better methods of sizing radiators and selecting boilers of the proper capacity.

This paper is intended to give figures by which the seasonal or monthly steam consumption can be calculated. For that reason it is not very important to attempt to take into account the minor things that affect heat consumption, such as wind velocity, the color of the building, and the wetness of the building, as was pointed out by two of the speakers.

The color of a building under consideration is, perhaps, unknown, or the exact wind velocity or direction near the building may not be known, but the ultimate aim of this study is to develop a simple formula which may be applicable to general use. In the district heating business such a method is needed every day, because we attempt to compare the customer's use of steam one month with another, so that we can warn him if the consumption becomes excessive.

Professor Rowley raised the point about the 65 F base. It seems to me that 65 F is a more logical temperature to use than 70 F, even if it were not borne out by experience. A building that is heated to 70 F, or a little above, in the day time is nearly always cooled at night, so that the average temperature throughout the 24 hours is likely to be more nearly 65 F than 70 F.

I cannot entirely vouch for the curve in Fig. 3 because it was taken from a report by R. M. McWhitty of St. Louis. In view of the fact that questions have been raised about it, it should be checked further. That is, I am not sure how many buildings it covers and whether the building for which the 70 F base is shown is the same building as the 65 F base.

In Fig. 4, concerning which Professor Rowley has some comments, I do not believe the minor differences between days 2 and 3 should be given very much consideration, because we could find 50 curves of pairs of days like this in which the minor differences might be in the opposite direction, but they would all show that a day in mid-winter, or even in spring, used more heat per degree day than a day in the fall.

No. 1011

TESTS OF THREE HEATING SYSTEMS IN AN INDUSTRIAL TYPE OF BUILDING

By G. L. LARSON,* D. W. NELSON,** AND JOHN JAMES*** (*MEMBERS*), MADISON, WIS.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Wisconsin

THE manner in which heat is distributed in a heated space is known to have considerable influence on the cost of heating. Maintenance of a uniform desired temperature in the occupied zone and a minimum temperature above that zone is conducive to efficient heating. The present program was undertaken to determine the effectiveness of heating with three different systems. The tests were conducted in a 71 by 182 ft one-story saw-tooth structure attached to the Mechanical Engineering Building and which is occupied by the Heating and Ventilating Laboratory. The side walls are of brick and one end wall is of brick and stone. It is considered to represent the usual industrial building of this type.

DESCRIPTION OF BUILDING

An aerial view of the Mechanical Engineering Building is shown in Fig. 1. The main structure is three stories in height and surrounds the Heating and Ventilating Laboratory on the west, north, and east. The influence of winds from these directions is, therefore, problematical. These surrounding walls also exert considerable influence on the sun radiation received. A plan and elevation of the Laboratory is shown in Fig. 2. It is attached to the main building at the north end and also at the southeast through the boiler room. There is a heated basement under one-half of section No. 3 at the south end. The remainder of the concrete floor rests on the ground.

Fig. 3 shows an interior view of the Laboratory. It is one large open space except for a partition located one-third of the distance from the north end and which extends one-half way across the width of the room. A 7 by 10 ft opening was made through this partition to allow free circulation of air.

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Presented at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by John James.

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TABLE 1. HEAT LOSSES AND SURFACES FOR THE HEATING AND VENTILATING LABORATORY

SURFACE		AREA	HEAT LOSS COEFF.	DIRECT RADIATION		UNIT HEATER	
				TEMP. DIFF.	HEAT LOSS B.T.U. HR.	TEMP. DIFF.	HEAT LOSS B.T.U. HR.
WALL	EAST & W.	5449 sq ft	0.385	86.0 F.	180,500	83.0	174,300
	SOUTH	726 "	0.259	82.1	15,480	81.1	15,290
GLASS	LOWER	630 "	1.13	82.7	39,750	81.4	88,800
	ROOF	2898 "	1.13	101.7	333,000	90.8	297,500
FLOOR	NORTH	10254 "	0.1473	15.0	22,650	15.0	22,650
	SOUTH	2050 "	0.482	20.0	19,880	20.0	19,880
ROOF		12796 "	0.386	99.9	494,000	89.5	442,000
LEAKAGE	WALL	6175 "	0.0756	86.0	40,200	83.0	38,770
	GLASS	190 ft.	1.94	82.7	30,500	81.4	30,000
	DOOR	73 ft.	3.90	80.0	22,200	80.0	22,200
TOTAL					1,216,900		1,120,080

Table 1 is a summary of the various heat loss items. The heat losses have been calculated using the temperature gradient of 2 per cent per foot for direct radiation and 1 per cent per foot for unit heater systems as recommended in THE A. S. H. V. E. GUIDE, 1934. This table shows that the roof and skylight heat losses are considerably over half of the total.

DESCRIPTION OF HEATING SYSTEMS

The Laboratory Building as originally built, which was previous to the construction of the remainder of the Mechanical Engineering Building, was heated by a two-pipe vacuum system using direct radiation. This consisted of wall radiation placed along the walls and in the trusses directly below the skylights. The location of these radiators is shown in Fig. 2. The total direct radiation surface in this system was 4431 sq ft at the time of test. Fifty-three per cent of this amount is located directly below the skylights.

For the purpose of these tests, seven suspended unit heaters were installed and were located 10 ft from the floor in the arrangement shown on Fig. 2. These heaters were loaned for the purpose of test by the manufacturers. The

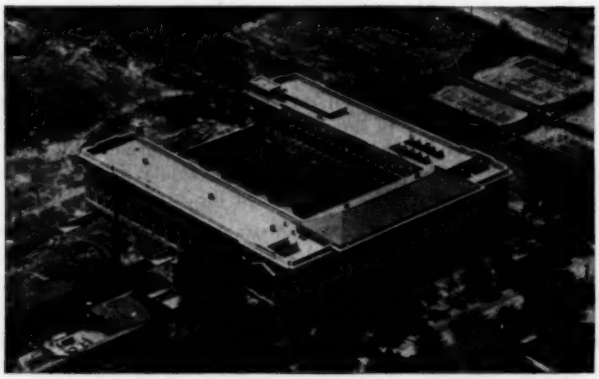

FIG. 1. AERIAL VIEW OF MECHANICAL ENGINEERING BUILDING AT THE UNIVERSITY OF WISCONSIN

TABLE 2. COMPARISON OF A. S. H. V. E. CODE TEST AND MANUFACTURERS CATALOG RATINGS FOR SUSPENDED TYPE UNIT HEATERS

HEATER NO	SPEED	B T U % DIFF	FINAL AIR TEMP ACTUAL DIFF	C.F.M % DIFF	WATTS PER 1000 C.F.M.
1	HIGH	+4.9	+9.7 F	- 5.9	105.2
1	LOW	+4.4	+4.5 F	- 3.0	
2	HIGH	+0.8	-6.2 F	+ 8.1	180.0
2	LOW	-4.5	-11.3 F	+ 8.9	
3	HIGH	+8.4	-8.7 F	+23.7	113.4
3	LOW	+18.3	-11.5 F	+33.9	
4	HIGH	-9.0	-10.9 F	+ 8.7	158.3
4	LOW	-7.2	-12.3 F	+10.4	
5	HIGH	+2.1	-4.5 F	+10.1	167.2
5	LOW	-5.6	-9.5 F	+ 9.4	
6	HIGH	+52.3	+5.4 F	+32.1	155.4
6	LOW	+38.9	-3.6 F	+51.6	
7	HIGH	+5.1	+0.4 F	+ 3.6	140.3
7	LOW	-2.8	-3.4 F	+4.4	

NOTE - PERCENTAGES ARE ON BASE OF TEST VALUES
+ SHOWS CATALOGUE RATING HIGHER THAN TEST RATING

total capacity as selected from the catalogs was 4676 sq ft with 218 F steam and 60 F entering air; the heaters were selected to have approximately equal capacities.

Each of these seven units was tested for steam and air capacity in a test apparatus. These rating tests were conducted during the year 1932-1933 and were made in accordance with the Standard Code for Testing and Rating Steam Unit Heaters adopted by the Society in January, 1930. The test set-up is shown

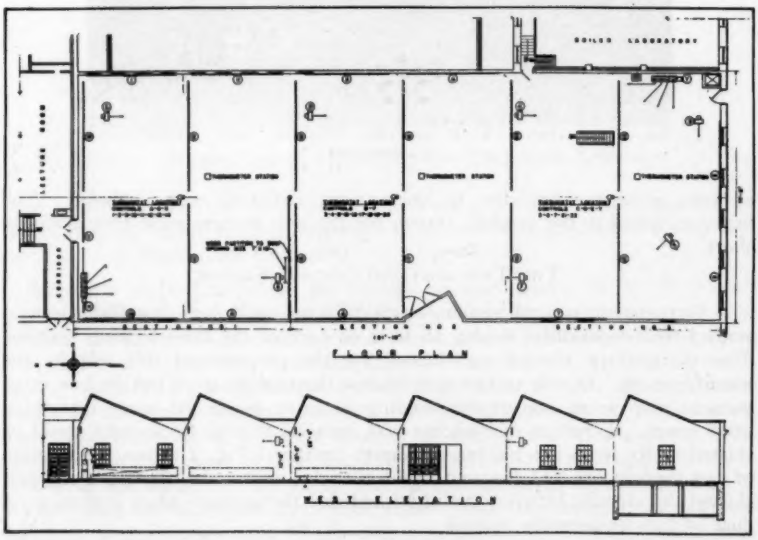

FIG. 2. FLOOR PLAN AND ELEVATION OF THE HEATING AND VENTILATING LABORATORY

in Fig. 4. The results of the rating tests are given in Table 2. The total heat capacity was found to be 1,036,700 Btu or 4318 sq ft equivalent direct radiation with 218 F steam and 60 F entering air. The total air moving capacity was 17,027 cfm. The total manufacturer's rating was 1,121,905 Btu and 20,265 cfm.[1]

Two floor unit heaters of identical size and construction were installed as a third heating system. Fig. 2 shows the location of these units and one of them appears in the photograph in Fig. 3. The total manufacturer's rating was 910,000 Btu or 3790 sq ft E.D.R. at 218 F steam and 60 F entering air.

The same mains and branches were filled with steam in each of the tests of the three heating systems. The piping was in excess for the unit heater

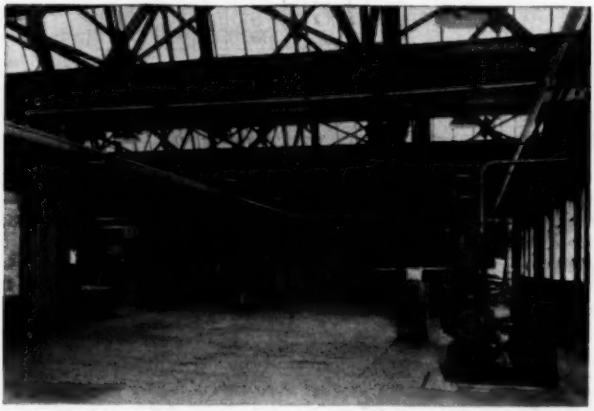

FIG. 3. INTERIOR VIEW OF THE HEATING AND VENTILATING LABORATORY

systems, since the branches to the various radiators were included. The branches added to the original system for the unit heaters were comparatively short.

THE THERMOSTATIC CONTROL SYSTEM

A thermostatic control system, electrically operated, was installed so as to control the temperatures during all tests on each of the three heating systems. This thermostatic control was loaned for the purposes of this test by the manufacturers. Double or two-temperature thermostats were installed on three posts on the center line of the building so as to divide the space into three equal zones. A system of switches was installed so that the control could be shifted easily from one heating system to another. Fig. 2 shows the location of the thermostats and the units controlled by each. An electric clock unit shifted the circuits between the night and day thermostats when operating on dual or two-temperature control.

[1] Acknowledgment is due to C. H. Novotny, as a graduate student, for his cooperation on the rating tests.

The thermostats opened and closed steam valves on the direct radiation, started and stopped the fan motors in the case of the suspended unit heaters only or in conjunction with the opening and closing of the steam valves, and for the floor unit heaters shifted the position of the by-pass damper. The fan motors on these floor units ran continuously. Since there were only two floor units placed at opposite ends of the Laboratory, only the two end thermostats were used. On certain tests of these floor units, thermostats were placed directly in front of the units and in the path of the return air.

METHOD OF TESTING

Steam consumption readings were taken daily at 5 p. m. on a calibrated condensation meter. Recording thermometers were installed in each of the

FIG. 4. GENERAL VIEW OF UNIT HEATER TEST CODE APPARATUS

three sections so that the sensitive elements were adjacent to heating units or in the path of the heated air from the units under test. By this means the *off* and *on* periods of heating were recorded.

Recorders were attached to the by-pass damper shafts of the floor unit heaters to show their position at all times. A recording thermometer was used to show the variations in temperature at the 5 ft level. This was moved about but usually was near the thermostat station in section No. 1. Fig. 5 shows typical charts of the temperatures at the 5 ft level for the three systems under dual control. Resistance thermometers were placed vertically at 4 ft intervals from floor to ceiling at a central location in each of the three zones as indicated on the floor plans of Fig. 2. The averages of the readings from floor to ceiling are shown for the three systems and the three sections on Fig. 6. As indicated on this figure each point represents a large number of readings taken on a great many days. Table 3 is a tabulation of all these readings to indicate the spread of temperatures from floor to ceiling in each section and for

each system. The averages used in constructing curves of Fig. 6 are shown at the bottom of this table.

It was necessary to correct the results to those obtainable on a standard day in order to make comparisons between the quantities of steam used with the various systems. The standard day was assumed as being one with a 39 F outside temperature. This was chosen as being the mean outside temperature for Madison during an average heating season. The corrections were made in direct proportion to temperature differences. The steam consumption was mul-

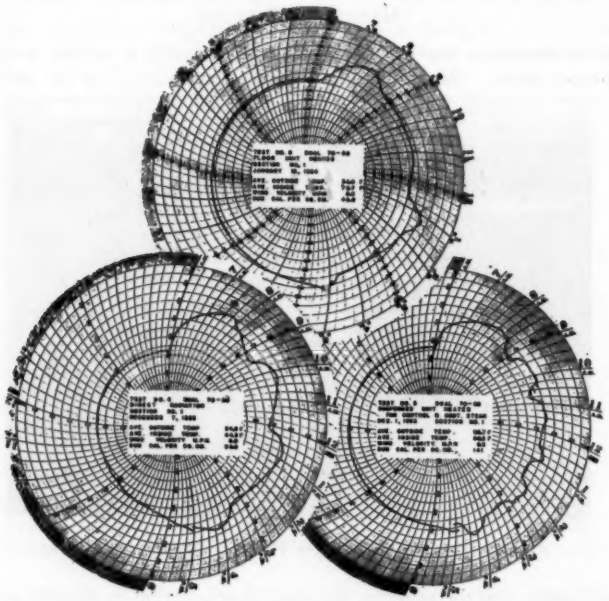

FIG. 5. TEMPERATURE RECORDER CHARTS FOR THREE HEATING SYSTEMS

tiplied by the ratio of difference between the inside temperature on a standard 39 F day and the outside temperature on the standard day over the inside temperature minus the outside temperature on the day of test. The inside temperatures for this purpose were taken at the 5 ft level. In securing the probable inside temperature that would exist on a standard 39 F day, the inside temperatures for all tests were plotted against the outside temperatures. From this it was determined that a change in outside temperature did not affect the inside temperature at the 5 ft level when continuous control was used. With dual control, it was found that lowering the outside temperature 10 deg lowered the inside temperature 1.9 F. This correction was accordingly made on dual tests to obtain the probable inside temperature on a standard day.

The weather conditions were obtained from reports of the local Weather

Bureau station located on the top of one of the campus buildings. Table 4 shows various factors in the weather for Madison over a long time period and for the season 1933-1934 in which these tests were made. This is included to make it possible to compare weather conditions of the tests with those in any other part of the country. In this table as well as in tables and figures showing results of tests, the sun radiation intensity and wind velocity are included, since they exert considerable influence.

RECORD OF TEST RESULTS

The results of the tests are shown graphically in Figs. 7 to 12, inclusive. In all of these the daily steam consumptions are plotted against the daily average

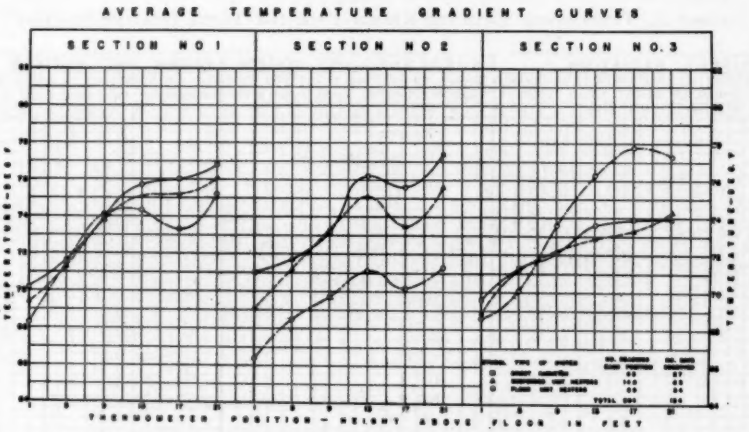

FIG. 6. TEMPERATURE GRADIENT CURVES FOR THREE HEATING SYSTEMS

outside temperatures. Each point shown is accordingly the result of a 24 hour test. The average results for each test period of from 5 to 9 days are shown as a cross with the identifying test number in a circle. These averages were corrected to a 39 F standard day by the previously mentioned method and indicated by circles at 39 F. The curves were plotted through this standard day point and with due attention to the averages of the tests concerned.

In general, the location of a daily point above the curve indicates a higher than average wind or a lower than average sun intensity, or both, and a location below, the reverse. Unfortunately, the wind velocities and sun intensities for individual days cannot be shown within the limits of this paper. The averages for test periods, however, are shown on these figures and in general wind and sun intensity differences explain the location above or below the curve.

Table 5 is a tabulation of the results of the three heating systems with continuous and dual control. The steam consumptions corrected to conditions of a standard day of 39 F are shown here for all tests. The differences in steam consumption are given in percentage values.

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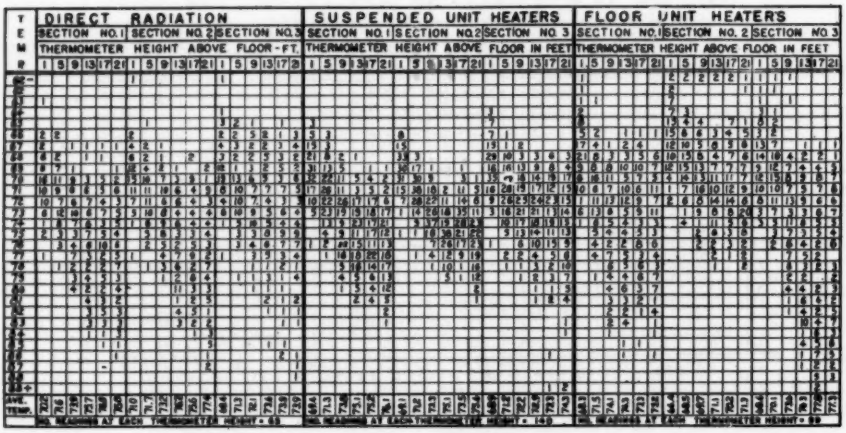
DUAL AND CONTINUOUS OPERATION OF SYSTEMS

The direct radiation system used 10,205 lb per day corrected to a standard 39 F day. When operated on dual control with the day thermostat set for 70 F for 11 hours and the night for 50 F for 13 hours, the consumption was 8700 lb per day. This is a saving for dual control of slightly less than 15 per cent. The wind velocity was 11.4 mph and the sun intensity was 380 cal per sq cm for the continuous tests. For the corresponding dual tests the wind velocity was 9.8 mph and the sun intensity was 121 cal per sq cm. These are in the direction to offset each other. It is considered that the sun intensity difference of 259 has a greater influence than the wind velocity difference of

TABLE 3. TEMPERATURE GRADIENT READINGS FOR THE THREE HEATING SYSTEMS



1.6. The difference between continuous and dual control is therefore probably somewhat greater than the 14.7 per cent shown in the table.

The tests on the suspended unit heater system with fan and steam control indicate a saving of almost 31 per cent for dual over continuous operation. The higher wind velocity and lower sun intensity during the dual tests over those of the continuous tests indicate that had these factors been equal in the two tests the saving would have been higher than 31 per cent.

With thermostatic control of the fan motor only, the steam being on the units continuously, a saving of 27.5 per cent is indicated for dual over continuous control. The wind velocities were equal. The sun intensity was slightly higher during the dual tests, which would indicate that the steam consumption should have been somewhat higher on the dual tests and the savings therefore somewhat less than the 27.5 per cent shown in the table.

As these results indicate, it would seem that the saving for dual control should be less when the steam is maintained on continuously day and night than when the dual control is applied to the control of steam as well as to the fan operation.

As mentioned previously, the 14.7 per. cent saving for dual operation of direct radiation is considered low because of the difference in sun intensities of 380 and 125 cal per sq cm. Had these sun intensities been equal, the saving might have been nearer those shown for the suspended unit heater systems. This saving for dual control with direct radiation also may have been reduced compared to those for unit heaters by the fact that lower outside temperatures existed during these tests over those for the unit heater test. This would cause steam to be on the radiators longer during the night period to maintain the set temperature, with consequent over runs in temperatures. This would

TABLE 4. MONTHLY WEATHER BUREAU AVERAGES AND MEANS FOR MADISON, WIS.

MONTH	MEAN TEMPERATURE		WIND				SUN INTENSITY CAL. PER SQ CM.	
			1869 – 1930		1933 – 1934			
	1869-1930	1933-34	AVE. VEL. M.P.H.	PREVAILING DIRECTION	AVE. VEL. M.P.H.	PREVAILING DIRECTION	NORMAL	1933-34
JANUARY	16.7	26.6	10.1	N.W.	9.8	N.W.	4,727	3,990
FEBRUARY	19.8	18.5	10.6	N.W.	9.7	N.W.	6,496	6,911
MARCH	31.0	29.9	11.3	N.W.	10.1	N.W.	9,820	9,627
APRIL	45.6	45.8	11.3	N.W.	11.1	N.W.	12,043	11,909
MAY-1934	57.6	58.7	9.9	S.	9.2	S.	14,706	16,934
JUNE-1933	67.1	76.0	9.2	S.W.	7.7	S.W.	15,585	17,333
JULY	72.2	73.6	7.7	S.W.	7.4	S.W.	16,243	16,009
AUGUST	69.9	69.6	7.6	S.	7.0	S.E.	13,892	15,291
SEPTEMBER	62.1	67.2	8.6	S.	8.8	S.W.	10,249	9,685
OCTOBER	49.9	49.0	9.9	S.	9.4	S.	7,102	7,725
NOVEMBER	35.0	34.0	11.0	N.W.	10.9	N.W.	4,323	4,326
DECEMBER	22.6	23.4	10.5	N.W.	9.4	N.W.	3,615	3,066

seem to cause the steam consumption on a standard day to be higher than it would have been had the outside temperature at the time of test been nearer to the standard day temperature of 39 F.

The saving of steam due to dual control when maintaining 60 F at night with the floor unit heaters was 16 per cent. The wind velocity was slightly lower on the dual test than on the continuous test. The sun intensity was considerably less, which probably more than offset the difference of wind velocities. It is likely, then, that the saving would have been greater than 16 per cent had the wind and sun factors been alike in the two tests.

With this same night temperature of 60 F, the suspended unit heaters showed 23.6 per cent saving. The sun intensity and wind velocity were both low in the dual test. However, the difference in sun intensities is greater, which indicates that the real saving would be larger.

Fig. 7 shows graphically the data discussed on the direct radiation system with dual and continuous control. Fig. 8 shows the corresponding data for the suspended unit heater system.

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TABLE 5. RESULTS FOR THE THREE HEATING SYSTEMS

TEST NO	PERIOD OF TEST	NO DAILY TESTS AVERAGED	SYSTEM	AVE INSIDE TEMP F	AVE OUTSIDE TEMP F	AVE WIND VEL M.P.H	SUN INTENSITY CAL PER SQ.CM.	AVE. TEMP DIFF IN - OUT	COR AVE IN. TEMP MINUS 39 F STD. DAY	ACTUAL STEAM CONSUMPTION AVE LBS PER DAY	AVE LBS. STEAM PER DEG T. DIFF. IN - OUT	COR STEAM CONSUMPTION AVE. LBS PER STD. 39 F DAY	SAVING IN PER CENT OF CONTINUOUS CONTROL	SAVING OVER DIRECT RADIATION	SAVING OVER SUSP. UNIT HEATER
CONTROL – CONTINUOUS 70 F – 24 HOURS														PER CENT OF TESTS 14-17	PER CENT OF TESTS 6-13
14	MARCH 16-22	7	DIRECT RADIATION	72.3	31.9	11.0	345	40.4	33.3	13,130	326	10,030			
17	APRIL 7-14	7	DIRECT RADIATION	73.3	43.4	11.8	415	29.9	34.3	8,350	280	9,580			
			AVE	72.8	37.7	11.4	380	35.1	33.8	10,740	306	10,205			
6	DEC. 21-29	8	SUSP U.H. FAN & STEAM CONTROL	71.7	10.0	7.9	127	61.7	32.7	17,810	289	9,440			
13	MARCH 7-14	6	SUSP U H FAN & STEAM CONTROL	73.8	24.6	10.2	337	49.2	34.8	13,650	277	9,640			
			AVE	72.7	16.3	8.9	217	56.4	33.7	16,030	285	9,520		8.72	
12	FEB. 26-MAR. 7	9	SUSP. U.H. FAN CONTROL-CONT. ST.	73.2	30.7	10.6	249	42.5	34.2	11,800	273	9,330			
16	APRIL 1-7	7	SUSP. U.H. FAN CONTROL-CONT ST	74.0	42.2	9.8	234	31.8	35.0	7,490	236	8,250			
			AVE	73.6	36.1	10.1	242	37.4	34.5	9,680	258	8,830		13.47	7.25
9	JAN 18-27	7	FLOOR U.H. THERM AT UNIT	71.6	33.7	11.1	143	37.9	32.6	11,210	296	9,660			
10	FEB. 5-13	6	FLOOR U.H. THERM. AT UNIT	71.9	18.7	10.1	227	53.2	32.9	16,320	290	9,550			
11	FEB 13-22	8	FLOOR U.H. THERM. AT COLUMN	72.0	23.8	9.8	257	48.2	33.0	12,280	257	8,790			
15	MARCH 22-31	8	FLOOR U.H DUAL THERM. AT COL.	73.5	28.8	9.2	304	44.7	34.5	12,430	266	9,180			
			AVE.	72.3	28.9	10.0	235	46.4	33.3	12,950	279	9,300		8.87	2.31
CONTROL – DUAL 70 F 11 HR. DAY – 50 F. 13 HR. NIGHT														PERCENT OF TESTS 1-4	PERCENT OF TESTS 2-5
1	NOV. 11-18	5	DIRECT RADIATION	63.3	21.8	10.9	161	41.5	27.6	13,450	324	9,840			
4	DEC 3-12	8	DIRECT RADIATION	63.9	28.9	9.1	96	35.0	27.4	11,580	312	8,560			
			AVE	63.7	24.3	9.8	121	39.4	27.5	12,510	317	8,700	14.73		
2	NOV. 18-25	7	SUSP U.H. FAN & ST CONTROL	65.4	37.6	9.9	139	27.8	26.7	6,575	236	6,300			
5	DEC 12-21	7	SUSP. U.H. FAN & ST. CONTROL	62.5	27.1	10.7	81	35.4	25.8	9,425	266	6,970			
			AVE.	64.0	32.3	10.3	110	31.7	26.2	8,000	252	6,590	30.78	24.25	
3	NOV. 25- DEC. 3	7	SUSP. U.H. FAN CONTROL-CONT ST.	65.3	37.3	9.8	128	28.2	26.8	7,750	275	7,375			
18	APRIL 15-25	9	SUSP U.H. FAN CONTROL-CONT. ST.	69.9	42.5	10.2	388	26.4	29.2	5,105	193	5,640			
			AVE.	67.5	40.3	10.1	278	27.2	28.2	6,265	231	6,400	27.82	26.45	2.89
CONTROL – DUAL 70 F 11 HR. DAY – 60 F 13 HR. NIGHT															
7	JAN. 4-11	7	SUSP. U.H. FAN & ST. CONTROL	66.3	28.0	7.6	66	38.3	29.4	9,470	248	7,270	23.64	16.44	– 10.32
8	JAN. 11-18	7	FLOOR U.H. DUAL THERM. AT COL.	67.3	26.4	9.3	139	40.9	30.7	10,410	255	7,810	18.02	10.24	– 18.51

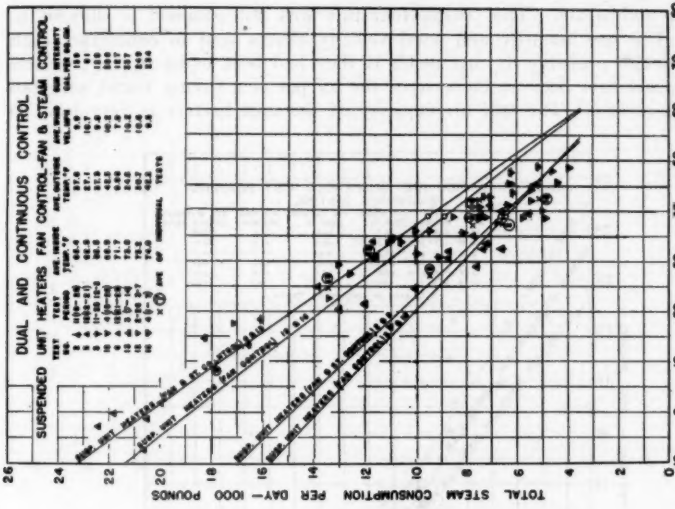
DUAL AND CONTINUOUS CONTROL
SUSPENDED UNIT HEATERS FAN CONTROL—FAN & STEAM CONTROL
TOTAL STEAM CONSUMPTION PER DAY — 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE — DEG. F

FIG. 8. STEAM CONSUMPTION CURVES FOR DUAL AND CONTINUOUS CONTROL WITH SUSPENDED UNIT HEATERS

Suspended unit heaters with continuous thermostatic control showed a saving in steam consumption of 6.7 to 13.5 per cent over direct radiation. The 6.7 per cent was secured in the tests with control of the steam supply as well as of the fan motors. The 13.5 per cent was found with control of the fan motors only. The greater saving would be expected with control of both air

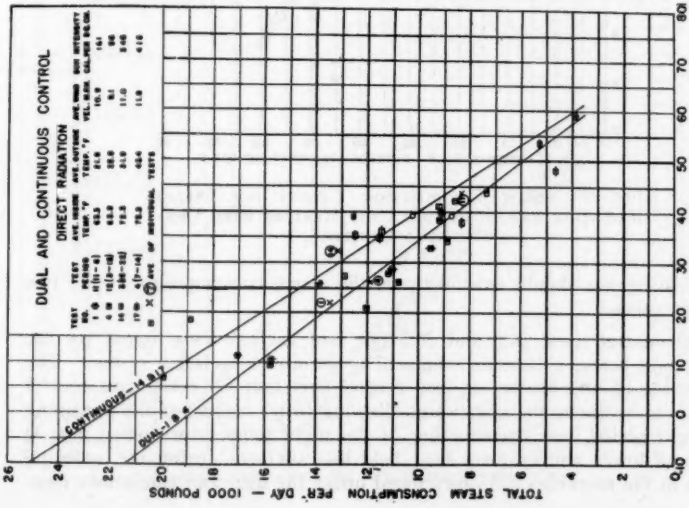

FIG. 7. STEAM CONSUMPTION CURVES FOR DUAL AND CONTINUOUS CONTROL WITH DIRECT RADIATION

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movement and steam. The reason that this was not obtained is difficult to explain. The sun intensity and wind velocity effects tend to counteract each other, although probably the net result if they had been equal in the two sets of tests would have been an increase of the 6.7 per cent saving found with fan and steam control. The still air capacity of the unit heaters is very low, so

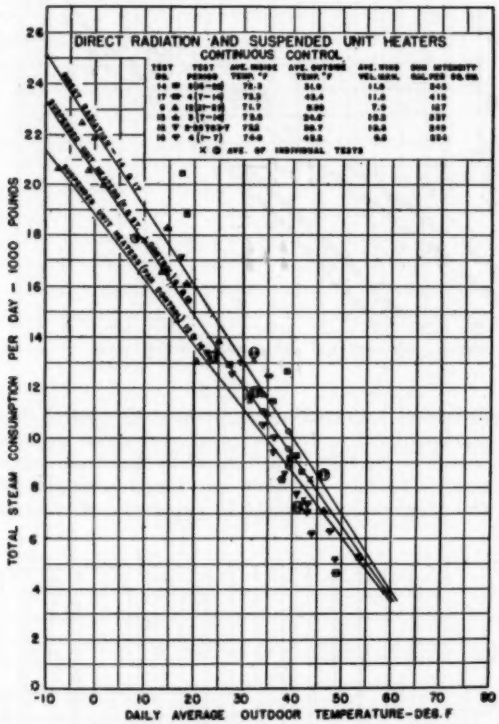

FIG. 9. STEAM CONSUMPTION CURVES FOR DIRECT RADIATION AND SUSPENDED UNIT HEATERS WITH CONTINUOUS CONTROL

that little difference should exist between the steam consumptions of the two methods of control.

On dual control tests, 24.3 and 26.5 per cent savings were found for the suspended unit heater system as compared to the direct radiation system. The greater saving of unit heaters on dual control over that for continuous control would seem to be due to the unit heater allowing faster cooling at the beginning of the night period, less overshooting of the night temperature when heat is required and lower temperatures near heat loss surfaces during the warming up period in the mornings. As mentioned under the dual and continuous com-

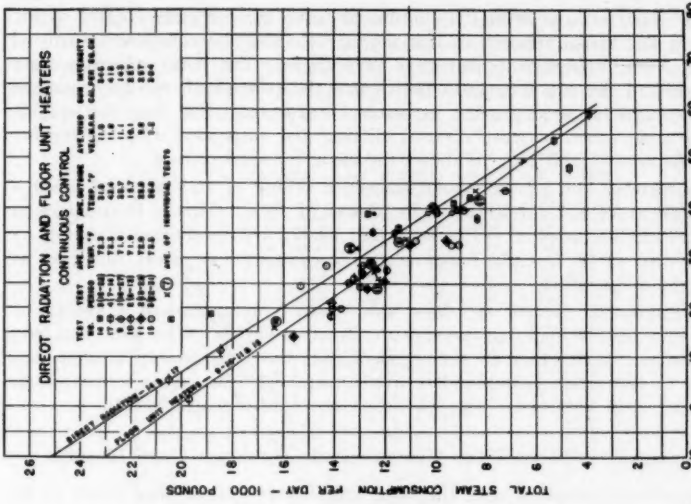
DIRECT RADIATION AND FLOOR UNIT HEATERS
CONTINUOUS CONTROL
TOTAL STEAM CONSUMPTION PER DAY – 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE-DEG. F

FIG. 11. STEAM CONSUMPTION CURVES FOR DIRECT RADIATION AND FLOOR UNIT HEATERS WITH CONTINUOUS CONTROL

parison for direct radiation, it is considered that the steam consumption for continuous control with direct radiation was low because of abnormally high sun intensity. This seems to indicate that the saving of unit heaters over direct radiation in this building is about 25 per cent for dual control and something over 13.5 per cent, possibly 20 per cent for continuous control.

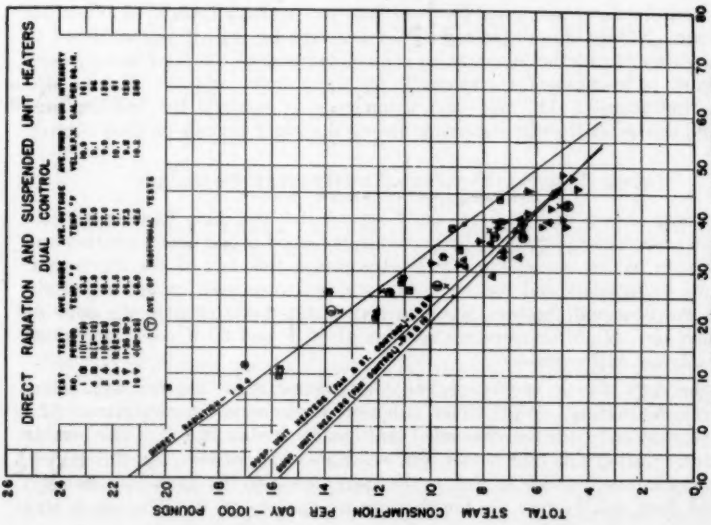
DIRECT RADIATION AND SUSPENDED UNIT HEATERS
DUAL CONTROL
TOTAL STEAM CONSUMPTION PER DAY – 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE-DEG. F

FIG. 10. STEAM CONSUMPTION CURVES FOR DIRECT RADIATION AND SUSPENDED UNIT HEATERS WITH DUAL CONTROL

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On the dual tests, as with the continuous tests, the complete control of air movement and steam resulted in less saving than the less complete control of air only. The difference, however, is very slight. The wind velocities were equal, but had the sun intensities been equal the percentage for the complete control would have been greater, as would be expected. Fig. 9 shows graphically the comparison between direct radiation and suspended unit heaters for continuous control and Fig. 10 shows the same for dual control.

The suspended unit heater system showed a saving of 16.5 per cent when a 60 F temperature was carried at night instead of 50 F. This is in comparison to direct radiation at 50 F. The floor unit heater system showed a saving of 10.2 per cent at 60 F night temperature, as compared to direct radiation at 50 F night temperature.

With continuous control the floor unit heater system showed a saving of 8.9 per cent over similar operation with direct radiation. The wind and sun effects balance each other to some extent, but the considerably greater sun intensity on the direct radiation test would make the real saving greater than the 8.9 per cent obtained. Fig. 11 shows these results graphically.

COMPARISON OF FLOOR UNIT HEATERS AND SUSPENDED UNIT HEATERS

The steam consumption with the floor unit heater system was found to be practically the same as that for the suspended unit heater system. On continuous thermostatic control, the floor unit heaters showed a saving of 2.3 per cent in steam consumption over the suspended unit heaters. On these tests the sun and wind effects do not differ greatly and tend to offset each other. Fig. 12 shows the data graphically for this comparison. A comparison on dual control with a night temperature of 60 F showed 7.4 per cent more steam used by the floor unit heater system than by the suspended unit heater system. The sun and wind effects are quite different but are in the direction of offsetting each other. Should this difference of 7.4 per cent be real, it is considered to be due to leakage past the temperature control dampers of the floor unit heaters. The steam is maintained continuously on these units and the motors also operate continuously. On mild days a tendency to overheat the building was noted and this probably was occurring during the night periods on dual control.

INSIDE TEMPERATURES AND TEMPERATURE GRADIENTS

The charts shown in Fig. 5 are daily records for the three heating systems for similar weather conditions. In each case the chart is for dual operation and was taken in section one at the 5 ft level within a few feet of the thermostat. The same thermostats and the same day settings were used except in certain tests of the floor unit heaters, where a thermostat was placed in the path of the return air. Night temperature settings of 50 F and 60 F were carried on tests involving dual control.

For the days shown in Fig. 5, the night temperature did not fall below the thermostat setting. With direct radiation a minimum temperature of 52.5 F was reached and with the suspended unit heaters it was 56.5 F. The outside temperature during this unit heater test was somewhat warmer, but this higher inside temperature was due in part to the steam being in the units continuously. With the floor unit heaters the temperature dropped to 66 F. The steam was

in the units continuously in these tests and the motors also ran continuously. This higher night temperature is considered to be due to leakage of air around the control dampers and conduction of heat from the steam coils to the air going through the by-pass passageway. The night temperature setting was 60 F in this case, but the record would appear the same had it been 50 F as in

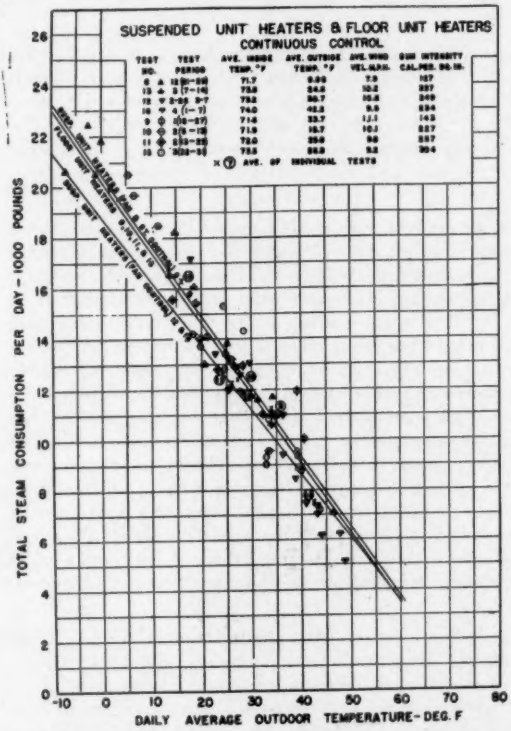

FIG. 12. STEAM CONSUMPTION CURVES FOR SUSPENDED UNIT HEATERS AND FLOOR UNIT HEATERS WITH CONTINUOUS CONTROL

the other two charts, since the temperature did not fall as low as the night setting.

The rate of cooling on the direct radiation system was 6.3 F per hour over the first two hours of night setting, which starts at 5 p. m. For the suspended unit heaters with steam on continuously, it was 3.3 F per hour with a slightly higher outside temperature. With floor unit heaters, the cooling rate from 5 to 7 p. m. was 4 F per hour.

The direct radiation system required one-half hour to bring the temperature to 70 F from the night temperature of 52.5 F. The temperature continued to

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rise until 80 F was reached. Three additional heating periods were observed during the day, the temperature varying from 68 F to 77 F.

The suspended unit heater system required one-half hour to raise the temperature from 56.6 F to 70 F in the morning. The maximum temperature obtained in this initial heating period was 77.5 F. The steam came on five additional periods during the day. The variation was from 68-69 F to 75.5-76.5 F.

With the floor unit heaters the temperature rise was delayed about 25 min beyond the time of shifting from night to day thermostats. About 30 min was required for the damper to change from the fully closed to the fully opened position. After a temperature rise was indicated, in 15 min the temperature reached 70 F from the night temperature of 66-66.5 F. There were two heating periods during the day and the variation in temperature was from 70.5 F to 82 F.

The temperature gradient curves shown on Fig. 6 are the averages of a large number of readings on all days of the tests. The average temperatures from floor to ceiling are shown for each section and for each heating system. The curves of each section are consistent with each other except as to the location of the curves showing temperature of the floor unit heater tests in sections 2 and 3. These units were controlled by thermostats in sections 1 and 3. The two units were about 190 ft apart. As a result the temperatures from floor to ceiling ran lower for this system in the middle section than in the two end sections. In section number 3, temperatures above the 5 ft level were high for the floor unit heater system. The vertical line of resistance thermometers was 30 ft from the unit in this section and the readings were undoubtedly influenced by the direct discharge from the unit. Consequently the averages shown are higher than the true averages over the entire section number 3. This curve does show that on the average the heated air rises considerably above the discharge outlets of the unit heater. The outlets are 8½ ft high, whereas the maximum temperature 30 ft from the outlet was found 17 ft above the floor.

In each of the three sections the average temperature 1 ft above the floor was higher with the direct radiation system than with the two systems using forced air circulation. In all sections the temperatures at the 5 ft level checked closely for the direct radiator and suspended unit heater systems. In section 1, all three checked closely at this level. In section 3, the temperature at the 5 ft level for the floor unit heater was low, just as it was high above this elevation.

In section 1, the average temperature at 21 ft was slightly less than 77 F for direct radiation and was about 1 deg less for the suspended unit heaters and 2 deg less for the floor unit heaters. In section 2, the direct radiation system showed an average temperature at the 21 ft level about 2 F higher than did suspended unit heaters. Had the 5 ft level temperatures coincided, the floor unit heater system would have had a slightly lower ceiling temperature in this section as compared to that of the others as it did in section 1.

The noticeable dips in the temperature gradient curves for the 17 ft levels in sections 1 and 2 are considered due to the influence of skylights with ventilating perimeters. Table 3 indicates the spread of temperatures for these temperature gradient observations.

CONCLUSIONS

The following conclusions may be drawn as applying to the three heating systems and the conditions under which the tests were conducted in the Heating and Ventilating Laboratory:

1. For dual control with the day thermostat set at 70 F for 11 hours and the night at 50 F for 13 hours, the saving of steam over continuous control with the thermostat set at 70 F for 24 hours was:

(*a*) For direct radiation 15 per cent and probably a much larger saving had sun intensities and wind velocities been equal;

(*b*) For suspended unit heaters with fan and steam control 30 per cent;

(*c*) For suspended unit heaters with fan control and continuous steam 25 per cent.

2. For dual control with the day thermostat set at 70 F for 11 hours and the night at 60 F for 13 hours, the saving of steam over continuous control with the thermostat set at 70 F for 24 hours was:

(*a*) For suspended unit heaters with fan and steam control 24 per cent;

(*b*) For floor unit heaters, with mixing damper control, continuous steam and continuously running motors 16 per cent.

3. The saving of suspended unit heaters over direct radiation both operating on continuous control was something over 13.5 per cent, and might have been 20 per cent had sun and wind effects been equal.

4. The saving of suspended unit heaters over direct radiation both operating on dual control (70-50 F) amounted to 25 per cent.

5. The saving of floor unit heaters over direct radiation both operating on continuous control was 9 per cent and would have been higher had sun and wind effects been equal.

6. The saving of steam of floor unit heaters over suspended unit heaters both operating on continuous control was 2.3 per cent.

7. The saving of steam of suspended unit heaters over floor unit heaters both operating on dual control (70-60 F) amounted to 7 per cent.

8. The average temperature gradient for all days of tests for the three heating systems over the entire period of test resulted in a change of two-thirds of one per cent per foot of height above the 5 ft level.

DISCUSSION

A. W. MOULDER (WRITTEN): I agree with, in a general way, conclusions 1—(a), (b) and (c)—Also conclusions 2—(a) and (b). These, however, will be varied by my comments on the remaining conclusions. The direction of savings would not be altered but probably would be increased in some instances.

The authors' conclusion 3 shows suspended unit heaters to consume 13.5 per cent to possibly 20 per cent less steam than direct radiation both on continuous control—the fans on the unit heaters only controlled. This I believe is a fair conclusion.

The authors' conclusion 4 shows a saving of 25 per cent in favor of suspended unit heaters over direct radiation when both were on dual control (70 F day, 50 F night). This seems reasonable.

The authors' conclusion 5 shows a preference of 9 per cent for floor type units over direct radiation (both on continuous control), but I am inclined to believe that this preference would not be as great if the floor type units had been doing the same job as the direct radiation. I will comment further on this later.

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The authors' conclusion 6 is based on an assumption rather than the actual result of tests.

The assumption was made by the authors that the steam consumption of suspended type units would be less when both steam and fan motors were controlled than it would be when motors only were controlled, whereas the actual test results show the reverse of this condition. With the type of construction involved in this case, namely, more than 50 per cent of the heat loss at the ceiling, it would seem entirely possible that the units when fan controlled only (steam on continuously) would consume less steam than were both the steam and fan controlled. This seems logical for two reasons:

(1) When the steam is controlled the temperature of the heating element drops from steam temperature to normal room temperature between each cycle. The increased load caused by re-heating these elements to steam temperature during each cycle would be greater, I believe, than the steam condensed with steam on and the fan idle. This is, of course, dependent upon the duration of periods between fan operations.

(2) When steam is shut off and the unit is doing absolutely no work even as a convector, the upper strata of air is apt to drop to a lower temperature in relation to the lower strata of air than it would if the steam was on continuously. If this were true the unit would condense more steam when it did start than it would had the upper strata remained at a relatively higher temperature. It must be borne in mind that this would probably only be true with this high ceiling heat loss and would not be true were this an intermediate floor, heated above, or if the roof was of such construction as to not present such a high relative heat loss.

In drawing conclusions the authors have in some cases based them on units which were fan controlled only and in other cases on units where both fan and steam were controlled. To be consistent I feel they should base their conclusions on the same conditions throughout. It would be more proper for them to make two sets of conclusions, one based on each condition, but I do not feel that they should be mixed.

Conclusion 6 shows the floor type unit 2.3 per cent more favorable than the suspended unit. This is misleading for the reason that as previously outlined they have based this value on the steam consumption of a suspended unit which was both steam and fan controlled, whereas the conclusion should have been based on the suspended unit heater which showed the most favorable, namely, the suspended unit heater installation which was fan controlled only. If this is done it will be found that the suspended unit heater, fan controlled only, will show a 5 per cent saving over the floor type unit instead of 2.3 per cent saving in favor of the floor type unit as indicated in this conclusion.

The authors' conclusion 7 is also misleading for the same reasons as outlined immediately above, namely, this is based on the suspended unit heater which is steam and fan controlled. If this was based on the suspended unit heater which is fan controlled only, the saving would have been still more favorable to the suspended unit heater than the authors have indicated. I cannot say just what this would be because the night-time temperature maintained with the floor type test as compared with the suspended type (fan controlled only) test was different and therefore not directly comparable.

The authors' conclusion 8 takes in the averages for all of the systems. It would be interesting if this were separated for each system.

By referring to Table 3 it will be noted that the thermometer readings for the suspended type unit installation are grouped much more closely than are the thermometer readings for either the direct or floor type unit installation.

The conclusions which the authors have reported would tend to favor the floor type units which in view of the previous discussion, we do not feel is justified. In

addition to this there is also decided evidence that the suspended heater accomplished a much better overall job than the floor type unit.

In the case of the floor type unit the middle or No. 2 section of the building was decidedly underheated. It seems fair to assume therefore that if this section had been heated to the proper temperature the floor type units would have consumed more steam.

By plotting an average temperature maintained in the three sections by the three heating systems, it is found that the temperature maintained by the floor type unit is 1½ F to 2 F lower throughout than the average maintained by the suspended unit.

In the case of Section No. 3 the floor type unit maintained extremely high temperatures, particularly at the higher levels. If the temperatures in this particular instance had been more nearly normal it would have naturally lowered the average maintained in the three sections by the floor type units proportionately, thereby increasing the difference of 1½ to 2 F to perhaps 2 to 2½ F. This variance may not seem of great importance, but when we recall that the standard day is considered 39 F outside, this 2 F difference represents about a 6½ per cent difference. It, therefore, seems logical that if the floor type units had maintained the same temperature as the suspended units the comparison of steam consumption would have been all the more favorable to the suspended units.

The outlets of the floor type units were 8½ ft from the floor and the suspended units were 10 ft from the floor. This fact is brought out because I believe that in reality for the suspended type unit to have done its best work it should have been considerably higher. It could have been mounted at least 5 ft higher than it was and, if this had been done, it is possible that the suspended unit would have shown still more favorable results.

While the type of building in which these tests were conducted no doubt represents a common industrial type, I feel that in this particular type, namely, where the ceiling heat loss represents such a high percentage of the total, the results of a comparative test of this sort would be quite different than in a multiple story building where there was heat above. The point is that with relatively high ceiling heat loss there is not the chance for heat banking at upper levels because of the fact that the heat is rapidly transmitted through the construction and therefore does not have an opportunity to build up.

It is also possible that if the suspended units were placed at as a high a level as practicable, and the fans allowed to operate continuously, but with steam automatically controlled, still greater economies in steam consumption would probably be effected.

Attention is also called to the fact that nowhere in this paper is any cognizance taken of the fact that suspended units generally being equipped with propeller type fans, require less electrical input in proportion to work done, than in the floor type multiple fan units.

A. B. ARNOLD[2] (WRITTEN): The authors of this paper are to be commended for the scope of the tests and the thoroughness of their report.

The fact that these tests were run under actual operating conditions with normal, full sized heating systems, makes them considerably more valuable, from the standpoint of manufacturers and purchasers of heating equipment, than small scale laboratory tests would have been.

These tests definitely show that there is a considerable variation in heating efficiency of the different systems in the particular building in question, but the extent to which these relative efficiencies can be applied to other buildings remains to be determined.

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[2] Asst. Chief Engr., Modine Mfg. Co., Racine, Wis.

The next logical step would seem to be a determination and evaluation of the factors which would cause the relative efficiencies of the different heating systems to vary with changes in building construction.

The writer would enumerate these factors as follows:

1. Wall temperatures as affected by temperature gradient, direction of heat delivery, and air motion adjacent to walls.
2. Relative proportions of radiated and convected heat in the output of the heating system.

The surprisingly small variation in temperature gradient would seem to eliminate this item and since the highest efficiency was obtained with unit heaters, the air motion resulting from their use cannot be extremely detrimental to heating efficiency.

This leaves direction of heat delivery and radiant heat as the two items which apparently have the greatest effect.

If direction of heat delivery is the principal factor affecting heating efficiency then results obtained on a building such as the one used for these tests should be generally applicable.

However, if direct radiant heat and the possibility of its passing through windows and walls is an important factor, then these results cannot be applied indiscriminately to any type of building.

A. E. SEELIG (WRITTEN): The work done by the authors of the paper in securing the comparative data and presenting their findings to the Society, constitutes a valuable contribution to the art and will, I trust, inspire more work of the same kind.

This is particularly true in view of the fact that the comparisons which they have made are limited to two distinct types of unit heaters (aside from the direct radiation used in the test) and any conclusions drawn from the observations by the authors can be held to apply only to the particular types of suspended unit heaters tested, to the locations of the units both in plan and elevation, to the number of units among which the heating load is divided, to the general arrangement of the individual units with respect to one another and the building contours, all as selected for this trial installation. In addition, the conclusions would hold only for a building of the general type used for the test. Thus, in a building with a flat roof instead of the saw-tooth construction, or again in a high structure with a monitor roof, it would be reasonable to expect quite different results.

It is obvious that the comparison between floor unit heaters and suspended unit heaters is inconclusive excepting within the limitations of the particular test layout and that a good deal of work remains to be done as an extension to what the authors of this paper have so ably started.

J. H. WALKER: This is a very valuable paper, but it is the kind that one cannot discuss without preliminary study.

One thing that impressed me in looking at the slides was the very wide variation in temperature for all three types of systems. It seems to me that temperature control was not as satisfactory as the thermostat manufacturers have indicated it should be.

Another thing that must be considered in comparing steam consumption is whether the temperature conditions were the same, or not, in all of the different studies. Obviously, if the weighted average temperature varies throughout the room, one will get a different steam consumption, and that may explain why one of these methods seems better than others.

I should like to ask Mr. James why the fans were run in the unit heaters during the night when no heat was required.

W. M. SAWDON: Were the outside temperatures the average of the maximum-minimum for the day, or were they the average of the hourly outside temperature?

D. S. BOYDEN: I didn't hear in the presentation any reference to any data being collected on the wind velocity and direction. Was that information tabulated?

C. C. TRUMP: I would like to ask about the sun. The photograph showed a black roof. If your standard day didn't take account of the standard sun condition, that big roof loss might amount to very much less if the sun happened to be shining on it.

It seems to me that your standard day ought to include a standard sun condition. The variation of temperature on the thermostats also seems to be large, and it would seem as if you could get thermostats which would provide closer control than those that are shown.

F. C. HOUGHTEN: Professor Larson and his co-workers are making a contribution to the industry in conducting these tests. I think we are all impressed by the tremendous number of variables involved.

There are several points that should be considered with respect to the floor-mounted units. In the first place, the lay-out of the building shows two unit heaters, yet the building is divided into three sections. The first section is divided half way across by a partition. The lay-out of the units as used would be all right for circulation, were that partition not there. It is true that there is a 7 x 10 ft opening, but that represents only about 15 per cent of the blocked-off area.

I believe that the arrangement would give better distribution of heat in the middle section had the north unit been at the other side of the building, so that it could have blown past that partition.

In other words, one unit is trying to care for one-third of the building, and the other unit for two-thirds, which accounts for some of the loss and the low temperature in the middle section.

Secondly, the controls were not functioning properly. Properly operating controls on a by-pass unit should allow that unit to float on the line and not run from a wide open to a tightly closed damper position. Mr. James emphasized that point when he pointed out that the thermostats in one case were located at the return of the unit, and in the other case in the center of the room. The sluggishness of response at the return accounts for the wide swings, and I think that the findings this year will indicate that the location of the thermostat in the heated zone will give better economy.

Another point that has already been raised is the question as to whether the units should have been operated during the night period. A by-pass unit is usually designed so that when the damper is wide open, there is still some heat capacity. Therefore, if you are trying to cool your building in the night and are operating for maximum economy, a unit of that type should be equipped with a limit control that would shut off when the damper is wide open.

I think the entire work deserves the highest commendation, and that it should be continued with a special study of improvement of the thermostatic control.

JOHN JAMES: In closing we would like to say that we appreciate that the results of these tests are not conclusive, but in view of the previous experiences and results, we are continuing these studies during the present year.

It is hoped that in the future we shall be able to present more conclusive test data on these various types of heating systems.

I believe Mr. Houghten has answered Mr. Walker's question concerning the operation of the units. The unit heater motors were operated to maintain continuous recirculation within the room, which was thought desirable at the time the test was conducted, but we shall continue tests in the future with the fan motors turned off at night when using dual operation.

The outside temperatures were determined from hourly temperature readings from the local U. S. Weather Bureau on a basis of mean temperatures. In Table 5,

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Results for the Three Heating Systems, values were given in condensed form for wind velocities and sun intensities which we attempted to correlate in analysing the conclusions and final results between the various systems. It is our intention this year to change the units which Mr. Houghten mentioned so that we may obtain better air circulation with the floor unit heaters and thereby increase the inside temperature existing in the center Section No. 2.

Professor Larson has some additional information to present on these tests that were conducted during the present year which indicate the temperature gradient advantages of an improved thermostatic control.

G. L. LARSON: I merely want to say that in a test of this kind there are a great many variables under which one has no control. The control equipment on the units was furnished us by the manufacturers, and it was installed according to their instructions. I think the manufacturers have learned a great deal from the test, as we have. It is possible to get better control by changing the type of control units, and I think the paper, from the curves that have been shown here, indicates that there is a great possibility of getting more uniform temperatures than these charts show.

This year we are continuing the tests with a change in the type of thermostat. I want to show three or four slides proving what can be obtained by other types of control than the kind we had available.

The recorders that we have are very sensitive, and a slight change in temperature is shown immediately upon the charts. There are temperature recorders of all kinds. You will very often see charts showing a line that can only be obtained by using a compass in drawing the chart. We have temperature recorders of that kind also. We could have used temperature records that would have shown absolutely perfect constant temperature lines. The recorders used are very sensitive and show what the manufacturers can do in the matter of temperature control with the present knowledge of the art.

We are going to continue the tests, and we hope to be able to get data from the results to help the Committee on Heat Requirements of Buildings to check up the known condensation, and the known conditions with the heat losses as calculated for the building.

No. 1012

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE FOR THE SUMMER OF 1934

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois.

The results presented in this paper were obtained in connection with the summer cooling investigation (1934) in the Research Residence,[1] Fig. 1, at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Special acknowledgment is due to the General Electric Co., the Utilities Research Commission of Chicago, Ill., and the Frigidaire Corp. for active cooperation in this investigation. Acknowledgment is also due to A. F. Hubbard, Graduate Assistant, for active participation in the detailed work of the investigation.

PREVIOUS investigations in summer cooling in the Research Residence during the summers of 1932 [2] and 1933 [3] made use of ice as the medium for cooling water circulated through coils placed in the forced-air heating system, and were confined largely to studies of the factors affecting the cooling

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[1] The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

[2] A. S. H. V. E. Research Paper entitled, Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 95.

[3] A. S. H. V. E. Research Paper entitled, Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 167.

Presented at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935, by A. P. Kratz.

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load, and to studies of the effectiveness of circulating air from the outdoors at night, both as a supplement to artificial cooling during the day and as a means for eliminating the necessity for artificial cooling during the day. In the course of these studies no outdoor air for ventilation was taken into the Residence during the periods when artificial cooling was required. The investigation for the summer of 1934 was undertaken to determine to what extent a two-ton mechanical refrigerating unit could be used to produce satisfactory cooling in the Research Residence, both when supplemented by the circulation

FIG. 1. VIEW OF RESEARCH RESIDENCE IN URBANA, ILLINOIS

of outdoor air through the second story at night, and when not so supplemented, under conditions in which approximately one air change per hour of outdoor air was used for the purpose of ventilation during the periods when artificial cooling was required.

DESCRIPTION OF THE RESEARCH RESIDENCE AND COOLING EQUIPMENT

The Research Residence, shown in Fig. 1, together with the forced-air heating system has been described in a previous paper.[4] For the purpose of this investigation, the Residence was equipped with awnings at all east, south, and west windows, and the sun parlor was isolated from the rest of the house by means of the doors opening into the dining room. The entire third story was regarded as an attic, and during the daytime was isolated from the rest of the house by means of a door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic during both day and night. With the exception of the space above the northwest bedroom and the small spaces adjacent to the dormer windows, the third story had hardwood floors

[4] Loc. Cit. See Note 2.

laid on pine sub-flooring. The small spaces adjacent to the dormer windows had no floors. In the space above the northwest bedroom 1 in. of insulating blanket was nailed to the upper edges of the floor joists. Hence, practically all second floor ceilings were at least equivalent to lath and plaster with flooring above it. No cooking was done in the kitchen, but the heat transmitted through the glass doors from the sun parlor, which was not ventilated by opening the windows, compensated for this to a certain extent. Unless otherwise specified,

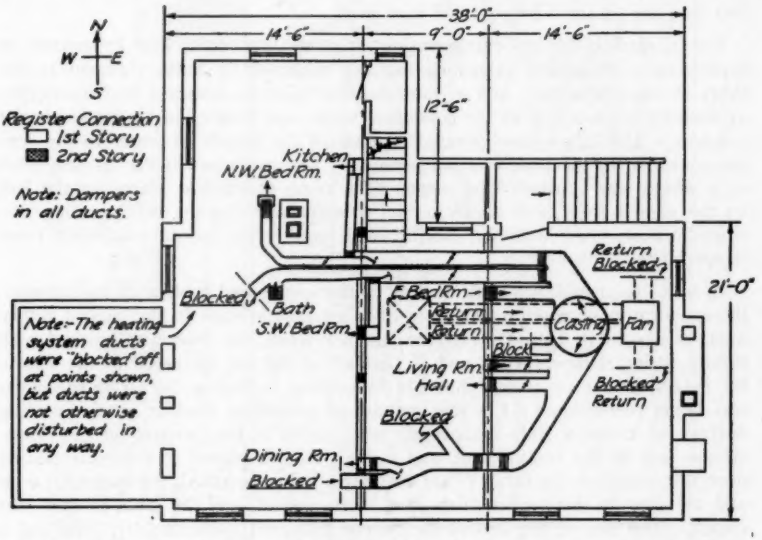

FIG. 2. BASEMENT PLAN SHOWING DUCT LAYOUT FOR FORCED-AIR SYSTEM IN USE FOR DISTRIBUTING COOL AIR

the state of the Residence was comparable for the work done during the three summers.

The arrangement of the forced-air heating plant and fan is shown in Fig. 2. For the purpose of this investigation, all return ducts with the exception of the central one containing the cooling coil were blocked. The delivery ducts to the sun parlor and third story were blocked, as indicated, and the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the cooling on these two stories.

The arrangement of the cooling plant is shown in Fig. 3. The condensing unit consisted of a double pipe condenser and a four-cylinder compressor driven by a 3 hp motor. This unit was self-contained. Cooling was accomplished by direct expansion of dichlorodifluoromethane in an evaporator unit placed in a by-pass in the central cold air return duct. The evaporator unit or cooling coil consisted of 32 rows of finned copper tubing placed 4 rows deep in the direction of air flow and 8 rows high measured along the vertical axis of the air duct. The

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tubes were ¾ in. outside diameter and had 6 fins per inch. The overall diameter including fins was 1½ in. The evaporator as installed in the duct was 17 in. high and 30 in. wide, and the gross or face area presented to the flow of air was 3.54 sq ft, while the net or free area was 2.18 sq ft. The nominal rating of the condenser and evaporator units as used was 21,700 Btu per hour with refrigerant temperature of 32 F, water temperature of 80 F, and ambient air temperature of 100 F, or 29,500 Btu per hour with refrigerant temperature of 32 F, and water temperature of 60 F, with an air velocity of approximately 350 fpm across the 3.54 sq ft of face area.

The operation of the refrigerating machine was controlled by means of thermostatic expansion valves having the thermostatic bulbs clamped to the tubes of the evaporator; and a water control valve so designed that the degree of opening for the flow of the condenser water was controlled by the condenser pressure. The latter also operated to shut off the supply of water to the condenser when the machine was not running. The operation of the cooling plant as a whole was controlled by means of a room thermostat placed in the hall on the second story, and which served to start and stop the refrigerating machine in accordance with the cooling load required to maintain constant room temperature.

In order to provide for both heating in the winter and cooling in the summer, the evaporator, or cooling unit, was installed in a by-pass in the central return duct, as shown in Fig. 3. For the summer work, the duct was blocked with tightly fitting dampers at *B* and *C*, and all of the air delivered to the fan in the forced-air system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, *E*, was placed in the by-pass on the downstream side of the cooling coil, and a door, *F*, was placed in the recirculating duct just ahead of the fan. When outdoor air was required, the basement door and the door in the recirculating duct were opened, and the slide damper was closed. The fan in the forced-air system delivered approximately 1300 cu ft of air per minute when recirculating the air in the house and 2200 cfm when using outdoor air at night. Outdoor air, for the purpose of ventilation during the periods when the cooling plant was operating, was provided by means of the duct shown as Detail *A* in Fig. 3. This duct contained a venturi section for the measurement of the volume of air delivered, and it was necessary to use a small fan in order to deliver the equivalent of approximately one air change per hour. Wet- and dry-bulb temperatures of the air and inlet and outlet temperatures of the condenser water were measured by means of thermocouples or thermometers placed at points indicated in Fig. 3.

METHOD OF CONDUCTING TESTS

During the summer from June 5 to September 16, continuous records were made, by means of temperature recorders, of the following air temperatures: outdoor, dining room, kitchen, first story hall, living room, east bedroom, southwest bedroom, northwest bedroom, and third story hall. Continuous records were also made of the wet- and dry-bulb temperatures of the air entering and leaving the cooling unit, and of the outdoor air taken in for ventilation. Other incidental air temperatures were observed at regular intervals. Relative hu-

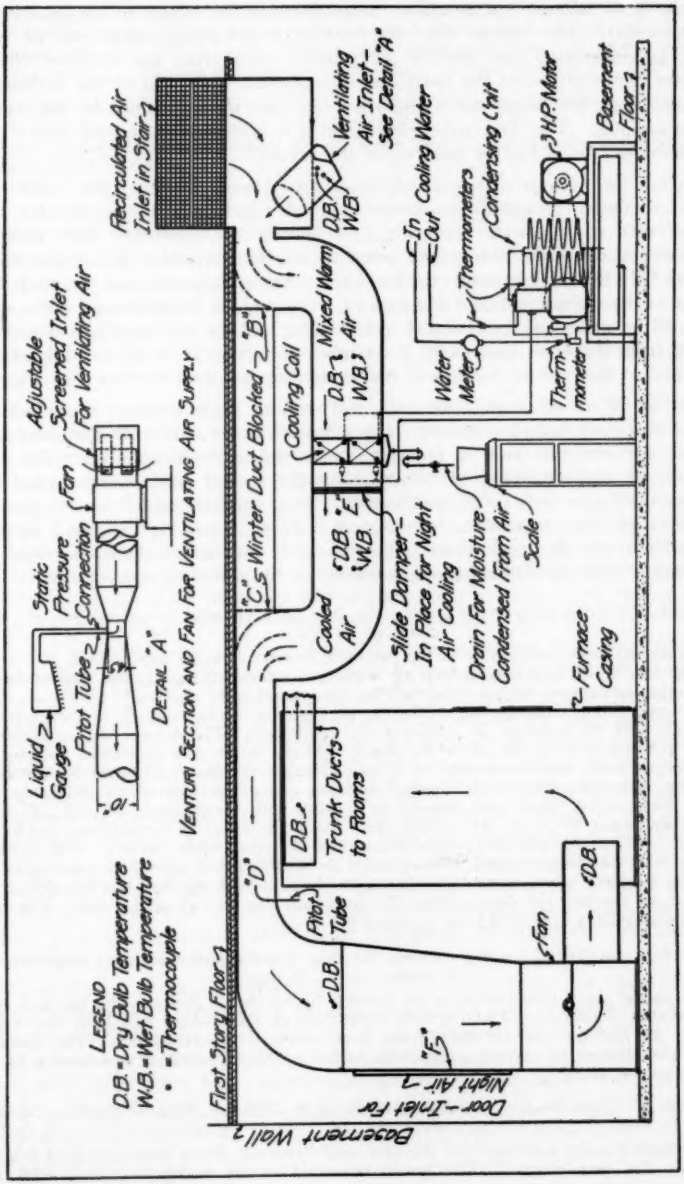

FIG. 3. DIAGRAM OF COOLING PLANT WITH MECHANICAL REFRIGERATION

midities both indoors and outdoors were observed by means of an aspirating psychrometer. The outdoor dry-bulb reading on this psychrometer served as a basis for correcting the outdoor temperature read from the recorder chart. Surface temperatures of the north, south, and west walls and of the northwest bedroom, east bedroom, and living room ceilings were obtained by means of thermocouples. The data were plotted on a continuous chart, and this chart was used as a basis for the analysis of the results.

During the periods of operation, observations were made of the weight of water circulated through the condenser and water jackets on the compressor, the temperature of the water entering and leaving the condenser and jackets, the temperature of the refrigerant entering and leaving and at the thermostatic control bulb in the evaporator, the head and suction pressures, and the electrical inputs to the compressor and fan motors. The weight of condenser water was obtained by means of a calibrated water meter and the air quantities were obtained from traverses made with Pitot tubes at section *D* in the central return duct and at the venturi section in the ventilating air duct as shown in Fig. 3.

During all of the tests, both with and without night cooling, the windows on the first story remained closed. The windows in the attic, with the exception of one opposite the door at the top of the stairs, remained open. For the purpose of night cooling, 11 windows on the second story were opened by raising the lower sash to the full extent. Two windows which were opposite registers and one window at the second floor stair landing remained closed. For all tests the Residence was operated strictly on the schedules described in connection with the following enumeration of the different test series:

Series 3-34. Artificial Cooling During the Day Supplemented by Circulation of Outdoor Air at Night

The second story windows and the attic and basement doors were closed at 7 a.m. and the fan, which had been delivering outdoor air through the system, was stopped. When the temperature of the indoor air on the second story rose to 81 F, the cooling plant was started, with the fan delivering through the cooling unit both the recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 4.5 recirculations of the air in the house, and the latter was equivalent to one air change per hour, making a total of 5.5 air changes per hour delivered by the fan. The fan was run continuously through both *on* and *off* periods of the refrigerating unit. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story until the effective temperature outdoors became equal to the effective temperature on the second story indoors. The refrigerating unit was then stopped; the second story windows and attic door were opened; and the dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system, the fan continuing to run until 7 a.m. The fan delivery was 2251 cfm or 9.5 air changes per hour.

Series 4-34. Artificial Cooling During the Day Not Supplemented by Circulation of Outdoor Air at Night

The second story windows were not opened during the 24 hours. The fan was run continuously during the 24 hours, delivering through the cooling unit both the recirculated air and the one air change per hour admitted for ventilation. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story.

Series 5-34. Circulation of Outdoor Air at Night Without Artificial Cooling During the Day

The second story windows and the attic and basement doors were closed at 6 a.m. and the fan was stopped. The house remained closed during the day, with no

attempt made to control the indoor temperature. The second story windows and the attic and basement doors were opened at 6 p.m., and the fan was started. During the night, the fan took air from outdoors and delivered it into the rooms through the duct system. The fan delivery was 2251 cfm, or 9.5 air changes per hour.

Series 6-34. Circulation of Outdoor Air at Night Without Artificial Cooling During the Day

Same as *Series 5-34* except that the windows were closed and the fan stopped at 7 a.m. instead of 6 a.m.

RESULTS OF TESTS

Cooling with Outdoor Air at Night

During the summer of 1933, studies were made [5] with various combinations of window openings and fans for cooling with outdoor air at night. With one exception these studies were made with both first and second story windows open. The one exception was made with a fan in the attic drawing in 3980 cfm which was approximately 33.6 air changes per hour based on the second story alone, or 16.8 based on the house as a whole.

Since it is undesirable in the average city residence to have the first story windows open all night, it seemed more practical in the case of the 1934 studies to confine the open windows to the second story alone. Furthermore, it also seemed more practical from the standpoint of the householder to employ the fan in the forced-air heating system circulating 2251 cfm or approximately 9.5 air changes per hour, rather than to install additional fans in order to circulate a greater volume of air. The effectiveness of night air cooling obtained under these conditions, as compared with the effectiveness of cooling for the basement fan with both first and second story windows open, and for the attic fan with the second story windows only open, is shown in Fig. 4.

In the previous paper [6] it was shown that the minimum indoor temperature attained at night was practically the same as that existing when the windows were closed at 6 a. m. Furthermore, it was shown that the effectiveness of different methods of night cooling could be compared by plotting the drop in indoor temperature from the time that the indoor and outdoor temperatures became identical to the time at which the indoor temperature attained a minimum, against the drop in outdoor temperature from the time at which the indoor and outdoor temperatures became identical to the time at which the outdoor temperature attained a minimum. These temperature drops are shown as *B* and *A* respectively in the insert in Fig. 4. In comparing different methods of night cooling, temperature drop curves giving greater slopes indicate greater effectiveness of cooling, and the maximum possible effectiveness would be represented by a 45 deg line, or one having a slope of 1.0 when the two temperature drops are plotted to the same scale.

From Fig. 4 it may be observed that in the case of the attic fan drawing approximately 33.6 air changes through the second story windows alone, the effectiveness in cooling the house as a whole, based on the average indoor temperature for both stories (Curve No. 2), was much less than that in cooling the second story alone based on the average indoor temperature for the second story (Curve No. 4). When cooling with outdoor air at night is used to supplement artificial cooling during the day, however, the effectiveness of night

[5] Loc. Cit. See Note 3.
[6] Loc. Cit. See Note 3.

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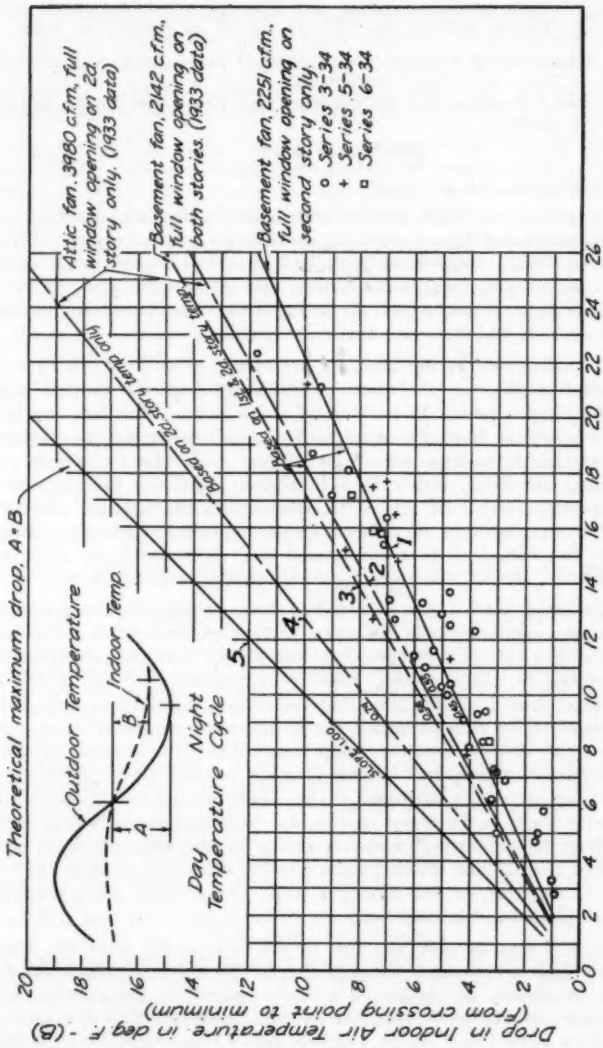
Theoretical maximum drop, A+B
Attic fan, 3980 c.f.m., full window opening on 2d. story only (1933 data)
Basement fan, 2142 c.f.m., full window opening on both stories (1933 data)
Basement fan, 2251 c.f.m., full window opening on second story only.
o Series 3-34
+ Series 5-34
□ Series 6-34
Outdoor Temperature
Indoor Temp.
Day
Night
Temperature Cycle
Drop in Outdoor Air Temperature in deg. F. -(A)
(From crossing point to minimum)
Drop in Indoor Air Temperature in deg F. -(B)
(From crossing point to minimum)

FIG. 4. TEMPERATURE DROPS FOR VARIOUS METHODS OF NIGHT AIR COOLING IN RESEARCH RESIDENCE

cooling in the house as a whole is of more significance than the effectiveness for the second story alone. The effectiveness of cooling the house as a whole was but slightly greater for the fan in the forced-air heating system circulating approximately 9.5 air changes per hour and with both first and second story windows open (Curve No. 3) than it was for the attic fan drawing in 33.6 air changes per hour through the second story windows alone (Curve No. 2). The effectiveness of cooling for the house as a whole with the fan in the forced-air heating system circulating 9.5 air changes per hour and the second story windows alone open (Curve No. 1) was somewhat less than that for either of the latter methods. The location of the points representing the test data indicates that the effectiveness of night cooling was not greatly influenced by whether the windows were closed at 6 a. m., as for *Series 5-34*, or at 7 a. m., as for *Series 6-34;* or whether they were opened at 6 p. m. or at the time that the effective temperature became the same indoors and outdoors, as for *Series 3-34.*

It has been mentioned that the slopes of the curves similar to the ones shown

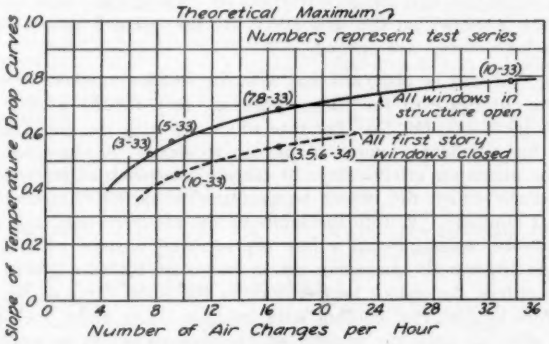

FIG. 5. INFLUENCE OF RATE OF CIRCULATION OF NIGHT AIR ON COOLING EFFECT IN THE RESEARCH RESIDENCE

in Fig. 4 afford a measure of the relative effectiveness of the different methods employed for cooling with outdoor air at night. Furthermore, the relative effectiveness of the different methods may be considered as dependent on the amount of air circulated and not inherent in the particular method used for circulating the air. Under these conditions, sufficient data are available from the studies of 1933 [7] and 1934 to establish the range of the number of air changes that would be required to prove effective for night cooling in a house of the type of the Research Residence.

The slopes of all of the curves obtained when the outdoor air was positively circulated by means of some type of fan are shown in Fig. 5, plotted against the corresponding amounts of air circulated expressed as a number of air changes per hour. Two curves were thus obtained. The upper curve represents conditions either in a two-story house in which windows are opened on both stories and the effectiveness of cooling is based on the average indoor tempera-

[7] Loc. Cit. See Note 3.

ture for both stories; or in the second story of a two-story house in which only the windows on the second story are opened and the effective cooling is based on the average indoor temperature of the second story alone. The fact that all of the points representing the data fell on a smooth curve, even though the amounts of window opening were not the same, indicates that the effectiveness of cooling for a given number of air changes per hour was more or less independent of the number and location of the windows opened, provided that the distribution of the openings was such as to give good distribution of the air. Furthermore, the fact that the point representing the cooling effect with 33.6 air changes per hour, obtained from consideration of the second story only, is on the curve, indicates that this type of curve is applicable to any space when the number of air changes per hour, the amount of window opening, and the indoor air temperature are considered in relation to the given space alone.

The lower curve in Fig. 5 represents conditions in a two-story house in which the windows on the second floor only are opened and the effectiveness of cooling is based on the house as a whole or on the average indoor temperature for both stories. It is evident that the effectiveness of cooling the house as a whole is less when the windows on the first story are not opened than it is when the windows on both stories are opened.

From Fig. 5 it may be observed that with the smaller numbers of air changes per hour the curvature of the curve decreases very rapidly as the number of air changes increases, but that the rate of decrease in the curvature becomes much less for the larger numbers of air changes. A slope of 1.0 would represent the maximum effectiveness of cooling, inasmuch as it would represent conditions under which the indoor temperature would be reduced to the same value as that outdoors. Within the limits of the observed data, the slope curve does not become asymptotic to a line representing a slope of 1.0, indicating that it would require air quantities far in excess of 33.6 air changes per hour in order to reduce the indoor temperature to the same value as that outdoors. It is therefore evident that cooling with outdoor air at night does not become reasonably effective until the number of air changes per hour reaches a value of approximately 9, and that above a value of 30 air changes per hour the gain resulting from increasing the amount of air circulated is very small. At some point the gain obtained by increasing the number of air changes per hour will probably be offset by the increased cost of electrical current required to operate the fan.

Cooling with Mechanical Refrigeration

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose a test made on June 27, 1934, was selected and the results are shown in Fig. 6 and Table 1. On this day the outdoor temperature was 97.1 F at 2 p. m. and reached a maximum of 99.0 F at 3:45 p. m. At that time the house was being operated on *Series 4-34* and the windows had remained closed during the night preceding the test. The fan in the forced-air system had continued to operate during the *off* period of the compressor, shown from 1:30 a. m. to 6:30 a. m. in Fig. 6, and ventilating air from the outdoors was drawn in, resulting in a rise in the relative humidity indoors. At 6:30 a. m. the compressor was started through the action of the

TABLE 1. TYPICAL OPERATING DATA AND RESULTS WITH MECHANICAL REFRIGERATION AT 2:00 P.M. ON JUNE 27, 1934[a]

1. Outdoor air	D.B. 97.1 F, W.B. 73.6 F, R.H. 33%
Moisture content, gr per lb dry air	86.8
2. Indoor air, avg. breath. level temp	1st Story 79.4 F, 2nd story 80.4 F
3. Indoor air, house avg. at breath. level	D.B. 79.9 F, W.B. 65.2 F, R.H. 45.5%
Moisture content, gr per lb dry air	70.0
4. Ventilating air	D.B. 94.3 F, W.B. 76.5 F, R.H. 44.5%
Moisture content, gr per lb dry air	108.6
5. Mixed air entering cooling coil	D.B. 81.5 F, W.B. 67.2 F, R.H. 47.5%
Moisture content, gr per lb dry air	76.9
6. Mixed air leaving cooling coil	D.B. 65.6 F, W.B. 59.9 F, R.H. 72.0%
Moisture content, gr per lb dry air	68.1
7. Air temp. drop through cooling coil, F	15.9
8. Temp. of cooled air leaving registers, 1st and 2nd story avg., F	70.0
9. Air temp. rise in ducts and casing, F	4.4
10. Basement air temp. at breath. level, F	77.3
11. Quantity of air circulated	1,303 cfm or 5,660 lb of dry air per hour
Density of air lb per cu ft	0.0724
12. No. of house air recirculations per hour	5.5
13. Cooling coil	face area 3.54 sq ft, free area 2.18 sq ft
Air face velocity, fpm	368
Air velocity through free area, fpm	598
14. Ventilating air	240 cfm or 1,015 lb dry air per hour
Density of air lb per cu ft	0.0707
15. Moisture condensed from air, lb per hour	7.11
16. Heat absorbed by cooling coil; total	29,065 Btu per hour
Heat due to moisture in air	7,466 Btu per hour; 25.7% of total heat absorbed
Sensible heat	21,599 Btu per hour; 74.3% of total heat absorbed
17. Water temp. through condenser, F	Inlet 59.7, Outlet 98.0, Rise 38.3
18. Water temp. through condenser, compressor and basement, F	Entering 59.7, Drain 100.0 , Rise 40.3
19. Quantity of condenser water, per hour	800.5 lb or 96.1 gal
20. Heat absorbed by water passing through condenser	30,659 Btu per hour
21. Heat absorbed by water passing through condenser, compressor and basement	32,262 Btu per hour
22. Heat equivalent of power input to compressor motor	9,980 Btu per hour
23. Net cooling load or heat gain on entire house (Item 21—Item 22)	22,282 Btu per hour
24. Ratio of net heat gain to total heat absorbed in cooling coil (Item 23/Item 16)	0.767
25. Temp. of cooling coil, F	Inlet 42.4, Control bulb 57.1, Outlet 74.0
26. Refrigerant pressures, lb per sq in	Suction 40, Discharge 130
27. Nominal rating of condensing unit with 32 F refrigerant, 60 F condensing water and 90 F ambient air	29,500 Btu per hour
28. Compressor motor data	Size 3 hp, Measured power rate 2.92 kw
29. Compressor speed	235 rpm
30. Recirculating fan data	Speed 568 rpm
Motor	Size ⅓ hp, Measured power rate 0.340 kw

[a] Test No. 7–34.

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thermostat and operated intermittently until 11:20 a. m. The compressor operated continuously at full load capacity from 11:20 a. m. to 11:15 p. m., at which time intermittent operation was resumed. During the long *off* period from 1:30 a. m. to 6:30 a. m., the cooling coil evidently warmed up and the temperature became practically the same as the air temperature. The intermittent periods of operation were not sufficiently long to establish equilibrium, and full capacity of the machine was not absorbed from the air until the compressor started to operate continuously.

During the hours from midnight to 5 a. m. the indoor temperature remained constant owing to the heat capacity of the structure while the calculated load curve showed decreasing cooling load. From 6 a. m. until 10 a. m. the actual average load approximated the calculated load. After 10 a. m., however, the actual load was less than the calculated load until approximately 6:10 p. m., at which time the two became equal. The calculated load attained a maximum of 40,300 Btu per hour while the actual load never rose above 30,500 Btu per hour. The effect of heat lag of the structure is most strikingly shown in the period from 6:10 p. m. to 11:15 p. m. At the latter time the calculated load had decreased to 12,500 Btu per hour while 27,500 Btu per hour were still being absorbed from the air. During the day the temperature of the air indoors rose from 79 F to 81 F. From 6:10 p. m. to 11:15 p. m. the indoor temperature was reduced from 81 F to 79 F. This was taken into account in the calculated load curve but is not sufficient to explain the wide discrepancy between the actual and calculated loads. From 11:15 p. m. to 3 a. m. the calculated load continued to decrease somewhat more rapidly than the actual load.

The calculated cooling load was based on the first and second stories only. It included the heat transmission through the walls, floors, ceilings, and glass, sun effect on walls, and glass, sensible heat brought in by the air used for ventilation, heat brought in by moisture in the air used for ventilation, heat from 4 occupants, heat from lights, and heat supplied by the electrical input to the fan, and was based on the hourly observed indoor and outdoor wet- and dry-bulb temperatures. The heat equivalent to the electrical input to the compressor motor was not included as it was regarded as all being dissipated in the basement. An attempt was made to determine the pressure built up in the first and second stories by the operation of the ventilating fan. While this did not result in a quantitative determination of the actual pressure, it did indicate that the pressure was slightly greater than atmospheric. Hence it was considered that infiltration through windows could be neglected and all infiltration could be credited to the air brought in for ventilation, amounting to approximately one air change per hour.

The sun effect based on unshaded windows was calculated by using the sun intensity curves for east, south and west vertical surfaces for the month of July from the work of F. C. Houghten [8] and others, and by multiplying the value thus obtained by 0.83 [9] to obtain the heat transmitted through the glass. Since all of the windows in the walls exposed to the sun were equipped with awnings, the heat transmitted through the shaded windows was obtained by

[8] A. S. H. V. E. Research Paper entitled, Heat Transmission as Influenced by Heat Capacity and Solor Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, pp. 231-279.

[9] Loc. Cit. See Note 2.

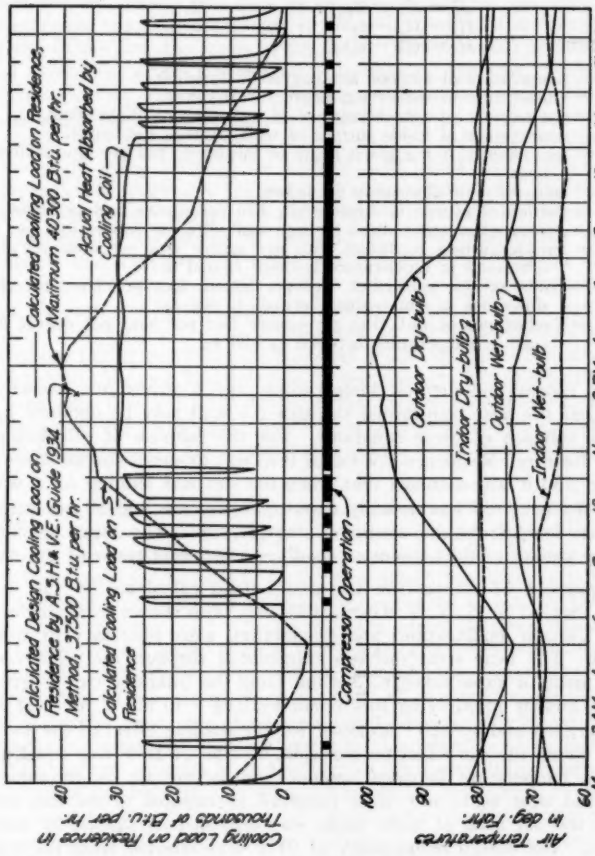
Calculated Design Cooling Load on Residence by A.S.H.&V.E. Guide 1934 Method, 37500 Btu per hr.
Calculated Cooling Load on Residence
Calculated Cooling Load on Residence, Maximum 40300 Btu per hr.
Actual Heat Absorbed by Cooling Coil
Compressor Operation
Outdoor Dry-bulb
Indoor Dry-bulb
Outdoor Wet-bulb
Indoor Wet-bulb
Time in hours
Cooling Load on Residence in Thousands of Btu. per hr.
Air Temperatures in deg. Fahr.

FIG. 6. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES, TEST NO. 7-34, JUNE 27, 1934; NO NIGHT AIR COOLING

taking 28 per cent of the heat calculated for unshaded windows, as indicated by more recent studies of F. C. Houghten [10] and others.

The heat transmitted through the walls due to the action of the sun on the exposed portions was calculated by the following method. Of the total intensity of the sun a part, I, depending on the angle of incidence, is received by the surface. A portion, eI, depending on the absorption coefficient, e, is absorbed by the surface. This tends to raise the temperature of the surface and of, eI, a portion, H_1, is lost by radiation and convection to the outdoor air, and another portion, H_2, is transmitted through the wall to the indoor air. From these considerations the following equations may be established: [11]

[10] A. S. H. V. E. Research Paper entitled, Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, C. Gutberlet and J. L. Blackshaw, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 101.

[11] Mechanical Equipment of Buildings, Vol. 1, second edition, by L. A. Harding and A. C. Willard, p. 248.

$$eI = H_1 + H_2, \text{ or } H_2 = eI - H_1$$
$$H_2 = (t_x - t_y)\, C = (t_y - t_i)\, f_i$$
$$H_1 = (t_x - t_o)\, f_o$$

In which t_o = temperature of outdoor air, degrees Fahrenheit.
t_i = temperature of indoor air, degrees Fahrenheit.
t_x = temperature of outside surface of wall, degrees Fahrenheit.
t_y = temperature of inside surface of wall, degrees Fahrenheit.
I = sun intensity for a given angle of incidence, Btu per square foot per hour.
e = emissivity, or absorption coefficient.
H_1 = portion of eI lost to outdoor air, Btu per square foot per hour.
H_2 = portion of eI transmitted through wall, Btu per square foot per hour.
f_o = outside surface coefficient, Btu per square foot per hour per degree difference in temperature between t_x and t_o.
f_i = inside surface coefficient, Btu per square foot per hour per degree difference in temperature between t_y and t_i.
C = conductance of wall, Btu per square foot per hour per degree difference in temperature between t_x and t_y.

For given indoor and outdoor temperatures and a given sun intensity the portion, H_2, of the heat transmitted through the wall may be obtained by the simultaneous solution of these equations. For the purpose of calculating the load for the Research Residence a value of 0.40 was assumed for the absorption coefficient, e, and it was assumed that when the sun was shining on a wall 50 per cent of the net area was actually exposed to the sun and 50 per cent was shaded by shadows from the awnings, shutters, etc. The latter assumption was based on studies of the amount of shading at different times of the day.

For the purpose of comparison the design load calculated by the method outlined in THE A. S. H. V. E. GUIDE 1934 has been shown as a single point in Fig. 6. Certain modifications and assumptions were necessary in applying this method. The heat from the sun transmitted through the windows was obtained by using a value taken at 2 p. m. from the intensity curves given in THE GUIDE, and by multiplying this result by 0.83 [12] to allow for heat intercepted by the glass and by 0.28 [13] to allow for the shading effect of the awnings. In order to obtain the sun effect on the walls a value of 25 F was added to the difference in temperature between outdoor and indoor air as recommended. The south and west walls only were regarded as exposed to the sun, and 50 per cent of the net area of these walls was regarded as shaded by awnings and shutters. The design temperature of 91 F was selected from the table in THE GUIDE and the average of 80 F was used for the indoor temperature. The temperature in the attic was assumed as 15 F higher than the outdoor temperature. It may be observed that this calculated design load was 37,500 Btu per hour as compared with the actual load of approximately 30,000 Btu per hour. The latter proved sufficient to cool the house satisfactorily.

The indoor relative humidity was approximately 61 per cent until 9 a. m., as indicated in Fig. 6. During the early period of intermittent operation of the compressor, the relative humidity decreased until the compressor operated continuously, at which time it assumed a value of approximately 45 per cent and remained practically constant at this value until the compressor again began to operate intermittently.

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[12] Loc. Cit. See Note 2.
[13] Loc. Cit. See Note 10.

Characteristic results obtained during the continuous operating periods of the compressor are shown in Table 1. The air conditions at different locations are given in the first ten items of the table. The dry-bulb temperature on the second story was only 1.0 F higher than that on the first, thus representing a very satisfactory balance of the cooling on the two stories. The ventilating air was taken from a large sunken window areaway on the north side of the Residence. This areaway was at the junction of the main body of the house and the north wing forming the kitchen, and hence was protected from the sun on both the south and the west. The temperature of the ventilating air was therefore somewhat lower and the relative humidity somewhat higher than the corresponding values for the outdoor air. The temperature of the cooled air leaving the registers was never lower than 68.0 F and no difficulty was experienced from drafts in the rooms. The rise in temperature of the air passing through the furnace casing and ducts was 4.4 F. The quantity of air circulated was 1303 cfm, which amounted to 5.5 recirculations per hour or 540 cfm per ton of refrigeration absorbed from the air. With this amount of air the velocity through the face area of the coil was 368 fpm and the temperature drop was 15.9 F. This corresponded closely with the 350 fpm and the 15 F drop used for the design. The free area velocity of 598 fpm did not prove to be sufficient to carry any condensed moisture away from the surfaces of the cooling coil.

The useful refrigerating effect, or heat absorbed by the cooling coil was 29,065 Btu per hour, which compared favorably with the nominal rating of 29,500 Btu per hour, although the temperature of the refrigerant as shown by Item 25 was considerably above the 32 F on which the rating was based. Item 25 indicates that considerable superheating occurred in a large portion of the evaporator or cooling coil. The ratio of the useful refrigerating effect to the heat absorbed by the water passing through the condenser was 0.95. This high ratio may be explained by the fact that considerable heat was lost from the condenser coil to the air in the basement, which would otherwise have been absorbed by the water and thus would have reduced the value of the ratio. However, this ratio, obtained for long operating periods of the compressor on each test, proved valuable in checking the calculated refrigerating effect for the periods of intermittent operation, during which the validity of using average temperatures for the air and average weights for the water condensed on the coil is somewhat uncertain. Such calculations based on averages for widely fluctuating conditions resulting from intermittent operation are always subject to some uncertainty.

As shown in Item 16 the useful refrigerating effect, or heat absorbed by the cooling coil, was obtained by computing the heat absorbed by the air, which was separated into the sensible heat given up by the dry air and the heat given up by the change in the moisture content of the air in passing through the cooling coil. In connection with the latter heat quantity, some difficulty was experienced in obtaining accurate readings of the wet-bulb temperatures as observed by means of recording thermocouples. The wicks used on these recording thermocouples became fouled in a short time, thus affecting the accuracy of the observed wet-bulb temperature. However, a sufficient number of accurate readings was obtained to prove that no great amount of variation occurred in the weight of air and in the entering and leaving wet- and dry-bulb

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temperatures for the air passing through the cooling coil during periods of compressor operation. The weighed amounts of water condensed on the coil were therefore accepted as being accurate, and the heat absorbed was obtained by multiplying this weight by a constant value representing the heat given by the change in moisture content of the air per pound of water vapor condensed. This latter included the latent heat and superheat in the water vapor condensed, and the change in the superheat in the water vapor remaining in the air after passing the cooling coil, and amounted to 1050 Btu per pound of vapor condensed. The weight of water vapor condensed varied from 12.5 to 3.4 pounds per hour based on the periods of continuous operation for the compressor, or from 7.2 to 1.2 based on the total length of the test period over which artificial cooling was required.

Item 21, the heat absorbed by the water passing through the condenser, the compressor heads, and the piping in the basement, H_w, represents all of the heat that was removed from the house with the exception of that which might have been lost through the basement walls. In order to separate the heat gains introduced by the presence of the refrigerating machine in the basement, such as electrical and mechanical losses in the compressor motor, mechanical losses in the compressor and drive, and heat losses from the condenser and compressor heads, from the net heat gains which were independent of the presence of the machine, the following heat balance on the house as a whole may be made:

$$H_w = H + H_e, \text{ or } H = H_w - H_e$$

In which H_w = Total heat removed by the cooling water from the point of entrance to the point of drainage.

H = Net heat gain of house, including heat transmission through walls, ceilings, and glass, infiltration, ventilating air, heat from lights and fan motor, and heat from occupants, but excluding heat lost through the basement walls.

H_e = Heat equivalent of electrical input to the compressor motor.

The net heat gain, H, is shown as Item 23 in Table 1. Item 16, the heat absorbed by the cooling coil, represents the useful refrigerating effect, and since ultimately all of the cooling in the house had to be accomplished by means of the air passing through the coil, it also represents the total amount of refrigeration that had to be expended to cool the house under the conditions of operation with the machine in the basement. If this heat is represented by H_r, then the ratio $\frac{H}{H_r}$ becomes an index of the additional cooling load imposed by the presence of the refrigerating machine in the basement, since this ratio would be 1.0 if it were possible to operate with no heat loss from the machine or condenser. In some respects this ratio is analogous to the overall house efficiency which may be obtained for winter heating. The value of 0.767, given as Item 24, is relatively low as compared with overall house efficiencies obtained for winter heating. Since $H = H_w - H_e$ is the net heat gain for the house, it may be observed that for given outdoor weather conditions any increase in the heat loss from the machine or condenser must be compensated for by readjustments in H_w and H_e. These readjustments must be such that the difference, H, remains constant. Any such increase in losses, however,

will be directly reflected in an increase in H_r. Hence the ratio $\frac{H}{H_r}$ will decrease rapidly as the losses from the machine or condenser are increased. Insofar as operating costs are concerned, it would therefore seem to be advantageous to insulate the condenser or to install the machine outside of the house.

The general results for all of the tests with mechanical refrigeration are shown in Table 2. It may be observed from Column 33 that beginning with test No. 17-34, a decrease in the ratio $\frac{H}{H_r}$ occurred up to, and including, test No. 24-34. This was caused by a gradual loss of the charge of refrigerant occurring over this period. As a result, the compressor did not operate as efficiently, and the heat losses from the machine and condenser became a larger portion of the useful refrigerating effect. The refrigerant had no odor and the gradual loss of the charge was obscured by other factors. A small decrease in the drop in temperature was observed, but it was erroneously attributed to an increase in outdoor relative humidity which occurred at about this time. Furthermore, the electrical input to the compressor motor which was observed by means of an integrating wattmeter did not decrease sufficiently to attract attention. The indoor temperature, however, was maintained lower than 81 F until test No. 22-34. Hence, only the three tests, Nos. 22-34 to 24-34, inclusive, have been regarded as of doubtful value, and these are useful in determining the effect on comfort of maintaining an indoor temperature higher than 81 F, which is discussed later. The machine was recharged after test No. 24-34 and on test No. 26-34, during which the maximum outdoor temperature rose to 101.4 F, the ratio $\frac{H}{H_r}$ rose to a normal value and the indoor temperature was maintained at 80.3 F, indicating that with a normal charge of refrigerant no difficulty would have been encountered in maintaining an indoor temperature of 80 F on the three preceding days for which the maximum outdoor temperature was from 100.0 F to 102.8 F. This experience, however, in which the loss of the charge was obscured by other factors even with the machine under constant observation, indicates that from the standpoint of the householder, it would be advantageous to introduce some odor, or other positive means of indicating a leak before the loss of the whole charge has occurred.

The total cooling load for the day, expressed as the total Btu absorbed from the air during the overall period of compressor operation, is shown in Fig. 7 plotted against the degree hours above 85 F per day. The base temperature of 85 F was selected because it was found that usually no cooling was required unless the outdoor temperature was approximately 85 F or above. Curves for both total heat and sensible heat are shown, and the difference between these curves represents the heat due to the change in the moisture content of the air. This moisture load varied from approximately 25 per cent to 30 per cent of the total load. One point, at 60 deg hours, deviates widely from the total heat curve but does not deviate very widely from the sensible heat curve. On this particular test, No. 29-34, the outdoor relative humidity was 83.5 per

cent, the highest recorded during the summer, and the load due to moisture was excessive.

A comparison between the results from *Series 4-34*, for which no cooling with outdoor air at night was employed, and those from *Series 3-34*, for which

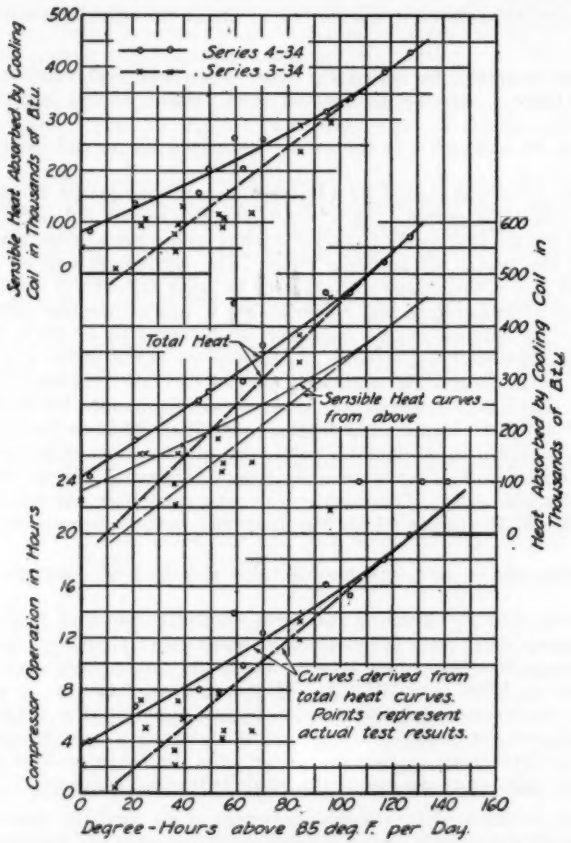

FIG. 7. HEAT ABSORBED BY COOLING COILS AND HOURS OF COMPRESSOR OPERATION PER DAY

supplementary night cooling was used, may be obtained from Fig. 7. The data for tests Nos. 22-34 to 24-34 inclusive have been omitted from these curves because the average indoor temperature was higher than 81 F and these tests were not regarded as being comparable with the others. The results from the two series of tests form two fairly well defined curves. More deviations occurred in the points representing the data from *Series 3-34* because the

amount of artificial cooling required during the day was directly influenced by the outdoor conditions on the preceding night. The curve represents median conditions, and if the night was exceptionally cool less artificial cooling was required the next day, while if the night was exceptionally warm the reverse was true. In the case of *Series 4-34,* for which the windows were not opened at night, the variations in the outdoor conditions at night were not so directly reflected in the amount of cooling required and hence less deviations occurred between the median curve and the points representing individual tests.

From the curves in Fig. 7 it is apparent that considerable saving in the amount of mechanical refrigeration required for cooling may be effected by supplementary cooling with outdoor air at night, even when the latter is confined to that which may be obtained by opening only the second story windows at night and employing 9.5 air changes per hour as furnished by the fan in the forced-air heating system. This is particularly true for milder weather. For *Series 3-34,* the windows were opened when the outdoor effective temperature became the same as that indoors. As the nights became warmer, this opening time occurred later, and when the degree-hours per day reached a value of approximately 110 no windows were opened and *Series 3-34* merged into *Series 4-34.* On days for which the maximum outdoor temperature reached 80 to 85 F the windows were opened at about 7 p. m. and no cooling was required the next day. When the maximum outdoor temperature was from 85 to 95 F the windows were opened sometime between 8 and 10 p. m. and artificial cooling was required the next day. When the maximum outdoor temperature was above 95 F and the indoor and outdoor effective temperatures did not equalize until later than midnight, the windows were not opened and artificial cooling was employed during the whole of the 24 hours.

The curves in the lower portion of Fig. 7 were computed from the total heat curves in the upper portion, and represent the total time that the compressor would be required to run delivering a constant mean refrigerating load of 29,000 Btu per hour. The actual running times obtained from the observed data for the individual tests have been plotted as points on the curves. A study of these points shows that, with the exception of four points near the upper end of the curves, they fell in about the same positions relative to the curves as the points fell relative to the total and sensible heat curves in the upper portion of Fig. 7. Of the exceptions, the three upper points represent data from tests Nos. 22-34 to 24-34 inclusive, which were excluded from the heat curves because an indoor temperature below 81 F was not maintained. The fourth point represents test No. 21-34, which was included in the upper curves because the indoor temperature was maintained at 80.7 F. From Column 33 in Table 2 it may be noted that for these four tests exceptionally low values of the ratio $\frac{H}{H_r}$ were obtained. A study of the points therefore indicates that when the refrigerant leak started with test No. 17-34, a gradual loss in efficiency occurred as evidenced by the reduction in the ratio of $\frac{H}{H_r}$, but the loss of part of the charge did not affect the output of the machine until test No. 21-34 was reached. At this point the output was decreased and although it was still

TABLE 2. COMPLETE DATA AND RESULTS FOR TESTS WITH MECHANICAL REFRIGERATION

Date 1934	Test No.[a]	Series No.	Weather Data: Degree Hours Above 85F	Degree Hours Above 90F	Sun, Hour Min.	Outdoor Air: Max. Temp. During Test, F	Outdoor Air: Min. Temp. Previous to Test, F	Outdoor Air: Average During Period Requiring Mechanical Refrig. Temp. F	Outdoor Air: Average During Period Requiring Mechanical Refrig. Hum. %	Indoor Air Conditions: At End of Night Cooling Temp. F	Indoor Air Conditions: At End of Night Cooling Hum. %	Indoor Air Conditions: Average During Period Requiring Mechanical Refrig. Temp. F	Indoor Air Conditions: Average During Period Requiring Mechanical Refrig. Hum. %	Temperature Difference: Outdoor Max. to Avg. Indoor, F	Temperature Difference: Average Outdoor to Avg. Indoor, F	Night Air Cooling, Previous to Test: Duration in Hours	Night Air Cooling, Previous to Test: Power Input to Fan, Kwhr
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
June 21	2-34	4-34	21.00	0	15:06	89.5	70.0	86.7	41.4	...	...	79.6	46.4	9.9	7.1	...	...
23	3-34	3-34	39.50	8.00	15:05	93.0	66.5	86.8	43.8	74.8	80.5	79.7	50.2	13.3	7.1	13.00	3.6
24	4-34	3-34	0	0	13:30	84.5	65.9	82.9	66.1	74.8	79.0	79.2	69.0	5.3	3.7	8.25	2.3
25	5-34	3-34	25.00	3.75	12:20	93.9	69.4	85.6	55.3	76.1	75.5	79.5	60.8	14.4	6.1	11.42	5.1
26	6-34	3, 4-34[b]	84.50	36.00	11:59	97.7	75.6	85.8	45.9	78.0	77.5	79.6	55.6	18.1	6.2	9.42	4.2
27	7-34	4-34	104.00	53.50	15:04	99.0	73.8	86.9	49.6	...	...	79.5	51.5	19.4	7.3	...	...
28	8-34	4-34	127.00	63.25	15:03	99.9	75.8	89.0	48.2	...	...	79.6	47.1	20.3	9.4	...	...
29	9-34	4, 3-34[c]	70.25	27.25	9:59	97.5	77.7	89.1	50.4	...	...	79.6	47.0	17.9	9.5	...	...
30	10-34	3-34	0.50	0	8:44	86.2	71.3	82.0	58.1	78.5	69.5	79.7	57.9	6.5	2.3	8.50	3.8
July 1	11-34	3-34	36.25	3.50	15:03	92.0	67.7	87.8	34.2	75.9	67.5	80.1	50.2	11.9	7.7	12.00	5.4
2	12-34	3-34	55.25	17.00	15:03	96.0	65.8	89.2	37.8	75.6	56.5	80.2	45.2	15.8	9.0	10.67	4.8
3	13-34	3-34	66.00	19.75	14:32	95.0	72.6	90.3	31.2	77.5	54.5	80.7	45.9	14.3	9.6	10.25	4.6
4	14-34	3-34	54.75	14.25	10:30	94.7	71.3	89.2	40.2	76.6	47.0	80.6	49.6	14.1	8.6	10.17	4.6
5	15-34	3, 4-34	84.50	31.50	9:00	97.0	75.2	86.9	53.2	78.5	66.5	80.4	54.7	16.6	6.5	9.00	4.0
6	16-34	4-34	3.75	0	5:25	87.9	68.6	84.1	52.1	...	...	80.4	55.0	7.5	3.7	...	...
10	17-34	3-34 Grav.	36.75	6.50	14:46	93.7	67.2	85.5	49.3	75.1	62.0	80.2	55.3	13.5	5.3	11.50	5.2
15	18-34	3-34	53.25	11.25	12:14	93.1	75.5	87.6	57.8	77.1	82.5	80.0	59.7	13.1	7.6	8.75	4.0
18	19-34	3-34	37.75	3.50	14:44	92.4	70.0	87.0	59.8	76.1	76.5	80.4	63.0	12.0	6.6	11.33	5.2
19	20-34	3-34	23.25	0	10:57	89.5	71.0	85.7	69.7	76.3	74.5	80.7	65.5	8.8	5.0	8.50	3.6
20	21-34	3, 4-34	96.25	42.25	14:41	97.8	76.7	88.0	58.6	79.4	74.5	81.2	57.2	16.6	6.8	8.25	3.7
21	22-34	4-34	131.75	73.75	14:40	102.0	78.2	88.2	64.2	...	...	82.2	56.7	19.8	6.0	...	...
22	23-34	4-34	107.50	53.75	14:38	100.0	77.2	87.0	61.7	...	...	82.6	55.6	17.4	4.4	...	...
23	24-34	4-34	141.25	85.25	14:36	102.8	72.8	87.8	55.2	...	...	83.5	53.0	19.3	4.3	...	...
25	26-34	3, 4-34	117.00	57.25	11:32	101.4	76.9	87.8	51.9	...	...	80.3	48.7	21.1	7.5	...	...
26	27-34	4, 3-34	49.50	8.25	8:25	94.1	71.5	88.3	51.6	...	...	80.2	50.4	13.9	8.1	...	...
Aug. 2	28-34	3-34	13.50	0.25	10:33	90.8	69.7	84.2	70.3	76.3	80.5	80.8	74.5	10.0	3.4	13.00	5.8
7	29-34	4-34	45.75	11.00	10:19	93.0	73.2	83.1	83.5	...	...	80.0	71.8	13.0	3.1	...	...
8	30-34	4-34	59.25	12.50	14:04	94.8	76.2	85.5	76.4	...	...	80.0	67.1	14.8	5.5	...	...
9	31-34	4-34	94.75	47.00	14:02	99.2	79.2	87.5	71.1	...	...	80.5	56.9	18.7	7.0	...	...
10	32-34	4-34	62.75	20.25	14:00	94.2	68.3	90.6	56.0	...	...	80.2	54.8	14.0	10.4	...	...

TABLE 2. (*Continued*)

Date 1934	Test No.	Series No.	Overall Test Period: Start of Compressor to Start of Night Air Cooling — Start	End	Duration, Hour	dCirculating Fan, Power Input During Overall Test Period, Kwhr	Compressor Operation: Time in Hours	Power Input, Kwhr	Condenser Water, Gal	Total Moisture Condensed, Lb	Heat Absorbed by Cooling Coil During Overall Test Period, Btu: Load Due to Moisture	Sensible	Total	Total Heat Absorbed by Cooling Water During Overall Test Period, Inlet to Drain, Btu	Heat Equiv. of Total Power Input to Compressor During Overall Test Period, Btu	Total Net Heat Gain on House, During Overall Test Period, Btu	Ratio of Net Heat Gain on House to Heat Absorbed by Cooling Coil
1	2	3	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33
June 21	2–34	4–34	9:00A	9:00A	24.00	5.3	6.67	18.9	514	41.27	43,333	136,444	179,777	188,181	64,534	123,647	0.688
23	3–34	3–34	2:32P	10:45P	8.22	2.2	5.73	16.4	482	37.33	39,197	131,728	170,925	174,700	55,998	118,702	0.694
24	4–34	3–34	5:04P	7:35P	2.52	0.6	0.83	2.3	....	6.98	7,330	....	....	....	7,853	....	....
25	5–34	3–34	12:55P	9:35P	8.67	2.8	5.00	14.6	486	44.67	46,904	106,129	153,033	168,900	49,852	119,048	0.778
26	6–34	3, 4–34[b]	11:17A	10:00A	22.72	7.4	11.92	34.5	1115	87.85	92,242	237,000	329,242	370,900	117,800	253,100	0.769
27	7–34	4–34	10:00A	9:00A	23.00	7.9	15.33	44.4	1519	116.14	121,947	340,868	462,815	505,100	151,604	353,496	0.764
28	8–34	4–34	9:00A	9:00A	24.00	8.0	19.87	56.6	1899	137.60	144,480	427,530	572,010	633,100	193,261	439,839	0.769
29	9–34	4, 3–34[c]	9:00A	10:30P	13.50	4.4	12.37	35.5	1180	95.87	100,663	261,237	361,900	396,900	121,215	275,385	0.761
30	10–34	3–34	11:00A	7:00P	8.00	2.6	2.20	6.2	213	11.48	12,054	51,674	63,728	70,100	21,170	48,930	0.768
July 1	11–34	3–34	2:57P	8:20P	5.38	1.8	3.25	9.4	302	15.73	16,516	76,832	93,348	102,700	32,096	70,604	0.756
2	12–34	3–34	2:20P	8:45P	6.42	2.1	4.83	13.8	432	24.92	26,166	107,588	133,754	147,000	47,120	99,880	0.747
3	13–34	3–34	11:40A	8:50P	9.17	3.0	4.78	13.5	427	16.15	16,958	117,658	134,616	148,800	46,096	102,704	0.763
4	14–34	3–34	1:58P	10:00P	8.03	2.6	4.22	12.0	376	17.78	18,670	98,923	117,593	129,700	40,974	88,726	0.755
5	15–34	3, 4–34	11:00A	9:00A	22.00	7.3	13.33	38.2	1206	89.60	94,080	288,005	382,085	414,200	130,434	283,766	0.743
6	16–34	4–34	9:00A	7:30A	22.50	7.5	4.00	11.3	355	26.22	27,530	82,840	110,370	121,900	38,584	83,316	0.755
10	17–34	3–34 Grav.	4:36P	10:00P	5.40	1.8	2.18	6.4	172	11.58	12,160	42,330	54,490	59,730	21,853	37,877	0.695
15	18–34	3–34	1:08P	10:15P	9.12	3.0	7.68	22.1	535	63.87	67,064	115,388	182,452	202,700	75,460	127,240	0.697
18	19–34	3–34	1:57P	10:30P	8.55	2.9	7.10	19.9	453	55.39	58,160	94,810	152,970	169,900	67,949	101,951	0.666
19	20–34	3–34	2:30P	10:45P	8.25	2.7	7.20	20.2	422	56.84	59,680	93,000	152,680	165,100	68,973	96,127	0.630
20	21–34	3, 4–34	9:11A	9:00A	23.82	7.8	21.82	61.0	1170	153.65	161,333	293,271	454,604	464,600	208,285	256,315	0.564
21	22–34	4–34	9:00A	9:00A	24.00	8.0	24.00	66.5	1160	150.12	157,626	300,156	457,782	471,900	227,064	244,836	0.535
22	23–34	4–34	9:00A	9:00A	24.00	7.7	24.00	65.9	990	119.69	125,675	278,109	403,784	414,000	225,016	188,984	0.468
23	24–34	4–34	9:00A	9:00A	24.00	7.8	24.00	64.6	808	89.19	93,650	257,153	350,803	348,200	220,577	127,623	0.364
25	26–34	3, 4–34	9:00A	9:00A	24.00	7.8	18.00	51.7	1728	124.89	131,135	391,933	523,068	562,000	176,530	385,470	0.737
26	27–34	4, 3–34	9:00A	9:15P	12.25	4.1	9.38	26.9	902	67.71	71,096	203,062	274,158	298,700	91,850	206,850	0.754
Aug. 2	28–34	3–34	4:50P	6:50P	2.00	0.7	0.50	1.5	48	4.03	4,230	11,027	15,257	16,400	5,122	11,278	0.739
7	29–34	4–34	9:00A	9:00A	24.00	8.0	8.00	23.5	791	94.56	99,288	157,302	256,590	276,800	80,241	196,559	0.766
8	30–34	4–34	9:00A	9:00A	24.00	8.0	13.87	40.6	1340	172.82	181,460	262,500	443,960	480,500	138,629	341,871	0.770
9	31–34	4–34	9:00A	9:00A	24.00	8.0	16.12	46.4	1369	141.49	148,565	315,235	463,800	501,500	158,433	343,067	0.740
10	32–34	4–34	9:00A	9:00A	24.00	7.9	9.83	28.2	872	85.42	89,690	203,587	293,277	315,800	96,289	219,511	0.748

[a] Tests Nos. 1, 25, 33 and 34 were omitted on account of incomplete data and unfavorable changes in the weather. [b] 3, 4 is a series No. 3–34 test. [c] 4, 3–34 is a series No. 4–34 test. [d] Circulating fan was run continuously during the overall test period.

possible to maintain an average indoor temperature below 81 F it was accomplished by an increase in the running time of the compressor. For the points representing tests Nos. 22-34 to 24-34 inclusive, it was not possible to maintain a temperature below 81 F even with the compressor operating continuously for 24 hours. Hence these three tests have been excluded in any consideration of the data. The point at 60 deg hours, representing test No. 29-34, is of interest as indicating the increase in running time that was made necessary by the moisture load imposed by an outdoor relative humidity of 83.5 per cent. It may also be observed that extrapolation of the curves to the 24 hour ordinate indicates that the compressor would not be required to operate continuously over the 24 hour period until the number of degree-hours per day above 85 F equalled, or exceeded, approximately 150. Since this condition does not occur at Urbana, Ill., the data indicate that a refrigerating machine of 2½ tons capacity is ample to maintain an indoor temperature not in excess of 81 F in a house similar to the Research Residence located where the climate is similar to that at Urbana, Ill.

On all of the tests in which the average indoor temperature for the test was not above 81 F the maximum indoor temperature did not rise above 81.5 F, and then only for a short time. With indoor relative humidities between 45 and 55 per cent this was equivalent to effective temperatures between 74 and 75 F. During the daytime these conditions were found to be comfortable both by members of the staff who occupied the Residence more or less constantly and by visitors who were present for short periods only. The maximum discomfort occurred with the *Series 3-34* tests during the hour just preceding the start of the compressor. After the windows were closed in the morning, the indoor dry-bulb temperature did not rise sufficiently to start the compressor until the late morning or early afternoon hours. During this time that the house was closed and the plant was not operating the indoor relative humidity increased and for a short time the house felt stuffy. Odors were also somewhat noticeable at this time. When the plant was started, both the slight discomfort and the odors disappeared. During the *Series 4-34* tests, for which the windows were not opened, the plant operated either continuously or at short intermittent periods over the whole 24 hours, and no discomfort or odors were noticeable. The admission of outdoor air for ventilation amounting to approximately one air change per hour was effective in overcoming any odors. At night, a temperature of 80 F was not entirely comfortable for a person lying down. The bed prevented the circulation of air over part of the body and the remaining surface was apparently not sufficient for effective cooling. It is therefore possible that sleeping quarters should be cooled to a somewhat lower temperature than living quarters.

On the three tests, Nos. 22-34 to 24-34 inclusive, for which the average indoor temperature was above 81 F, the maximum temperatures attained during the day were 84.5, 85.2, and 86.0 F. During these times the house was distinctly uncomfortable to the occupants. On coming in from outdoors an observer would be satisfied for a short time owing to the contrast, particularly if the outdoor temperature was approximately 100 F. This effect was not lasting, however, and indoor temperatures as high as 85 F do not seem desirable unless relative humidities lower than those obtained at the Research Residence are readily attainable.

Table 3 gives a summary of total quantities obtained for the season of 1934. The total of 2657.5 deg hours above 85 F, Item 3, indicates that this season was severe as compared with those for 1933 and 1932, for which the degree hours above 85 F were 2309.3 and 1470.7 respectively. This was caused by the comparatively large number of days on which the outdoor temperature rose above 90 F. The hours above 85 F for the seasons of 1934, 1933, and 1932 were 481.5, 493.2, and 329.1 respectively, while those above 90 F were 224.1, 208.4, and 121.8.

It is of interest to note that the total heat absorbed during the season with the two methods of operating the Residence from June 20 to September 16

TABLE 3. SUMMARY OF RESULTS OF TESTS WITH MECHANICAL REFRIGERATION AND NIGHT AIR COOLING FOR ENTIRE SEASON OF 1934

1	Total hours above 85 F for season of 1934	481.5
2	Total hours above 90 F for season of 1934	224.1
3	Total degree-hours above 85 F for season of 1934	2,657.5
4	Total degree-hours above 90 F for season of 1934	948.7
5	Number of tests with mechanical refrigeration	33
6	Total running time for fan during night cooling, hours	692.3
7	Average rate of power input to fan during night cooling, watts	410
8	Total power input to fan during night cooling, Kwhr	283.8
9	Total running time for fan during test period, hours	559.2
10	Average rate of power input to fan during test period, watts	314
11	Total power input to fan during test period, Kwhr	175.5
12	Total running time for fan including night cooling, hours	1,251.5
13	Total power input to fan including night cooling, Kwhr	459.3
14	Total running time for compressor, hours	335.7
15	Average rate of power input to compressor, watts	2,836
16	Total power input to compressor, Kwhr	952.1
17	Total power input to compressor and fan, including night cooling, Kwhr	1,411.4
18	Total quantity of cooling water, gallons	25,741
19	Total heat absorbed by cooling coil during season, Btu	8,354,881
20	Equivalent ice meltage during season, tons	29.0
21	Corrected ice meltage during season assuming water to the drain of 40 F, tons	27.5

was 8,354,881 Btu, which was equivalent to an ice meltage of 29.0 tons. It is not usual to reject the water to the drain at a temperature of 32 F in the case of an ice plant. On the assumption that the water is rejected at approximately 40 F and that advantage is taken of this additional cooling effect, an equivalent ice meltage of 27.5 tons would be obtained as shown by Item 21. This equivalent tonnage would have been slightly increased if the plant had been in operation about May 15. However, the few hot days in the latter part of May and the early part of June were accompanied by cool nights, and by employing night cooling with outdoor air, it would not have been necessary to start the plant during this time with the exception of not more than five days. The 27.5 tons may, therefore, be regarded as representative of the whole season's requirements under the existing conditions of operation. This figure is *not*

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comparable with the 43.3 tons of ice actually used during the summer of 1932. For the 1932 season the windows remained closed during the whole period from June 1 to October 1, and no advantage was taken of the possible supplementary cooling with outdoor air at night. Furthermore, for one-half of the season no awnings were used on the windows. For the work in 1934, full advantage was taken of cooling with outdoor air at night for a large part of the season and the Residence was equipped with awnings during the whole season. The 27.5 tons is to a certain extent comparable to the estimated 19 tons for the season of 1933 obtained under conditions of night cooling with the fan in the forced-air heating system and windows on both stories opened; if it is recognized that the season of 1934 was more severe than that of 1933, that the maximum amount of ice used per day was limited to 700 lb during the season of 1933, and that night cooling with the windows opened on both stories during the season of 1933 was somewhat more effective than similar night cooling with second story windows only opened during the season of 1934.

A discussion of comparative costs is not conclusive unless unavoidable differences in existing conditions are fully recognized. However, the total cost for electricity and water for the season's operation under local conditions is of some interest. The total electrical input to the compressor and fan, including the input to the fan during night cooling, as shown by Item 17, was 1411.4 kwhr. Therefore at the local rate of 0.031 per kwhr the cost for electrical current was $43.75. The total amount of water used, as shown by Item 18, was 25,741 gal. At the local rate of $0.37 per thousand gallons the total cost for water was $9.52. Hence, the total cost for electricity and water under the conditions of operation was $53.27. As previously mentioned, if the cooling plant had been operated from May 15 to September 16 instead of from June 20 to September 16, this figure would have to be slightly increased. However, the season would not have been increased by more than five days on which cooling was necessary and the plant would have had to operate during only a small portion of each day. On the other hand, if supplementary cooling with outdoor air had been used during the whole season a reduction in this cost would have been probable. Therefore, it seems reasonable to regard $60.00 as the approximate cost for electricity and water for the season of 1934.

CONCLUSIONS

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted:

(1) An indoor temperature of approximately 80 F with relative humidity below 55 per cent results in satisfactory comfort conditions in the living quarters of a residence. For complete comfort in sleeping quarters a somewhat lower temperature is desirable.

(2) The introduction of approximately one air change per hour of outdoor air for the purpose of ventilation is sufficient to prevent objectionable odors.

(3) A mechanical refrigeration unit capable of producing 2½ tons of refrigeration is sufficient to maintain conditions of comfort on two stories of a residence similar to the Research Residence when the outdoor temperature does not exceed 103 F, and an amount of outdoor air equivalent to one air change per hour is used for the purpose of ventilation.

(4) The use of the fan in the forced-air heating system to circulate outdoor air at night amounting to 9.5 air changes per hour is not as effective in cooling the Residence as a whole when the second story windows only are opened as it is when windows on both stories are opened.

(5) The cooling load during the daytime can be reduced by supplementing artificial cooling during the day with cooling by means of outdoor air circulated at night.

(6) Satisfactory cooling with outdoor air at night probably cannot be accomplished by employing less than 9 air changes per hour. The gain arising from the use of more than 30 air changes per hour is very small.

DISCUSSION

G. B. HELMRICH [14] (WRITTEN): Two years ago The Detroit Edison Company sponsored the installation of cooling equipment in a residence located in a Detroit suburb. The total space cooled consisted of 9 rooms with a total volume of 19,200

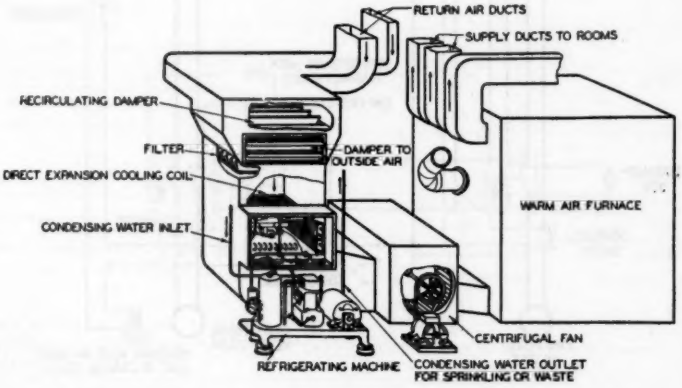

FIG. A. GENERAL ARRANGEMENT OF COOLING EQUIPMENT

cu ft. The residence was described in a paper [15] which was presented at the Annual Meeting of the Society in January 1933. During the summers of 1932 and 1933 cooling was accomplished by the use of ice and the operating results were discussed in this paper and in a discussion [16] presented at the Annual Meeting of the Society in 1934. The ice equipment was replaced by a refrigerating machine and direct expansion cooling coil in June 1934, and this equipment was operated during the summer of 1934.

The cooling equipment consists of a 2-ton refrigerating unit supplying dichlorodifluoromethane to a direct expansion forced convection cooling coil. The coil is placed in the main return air chamber of a conventional forced warm air heating

[14] Detroit Edison Co.
[15] Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.
[16] Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 195.

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system and no changes were made to the existing duct work except relatively minor ones to accommodate the coil. The general arrangement is shown in Fig. A. The operation of the machine is controlled by a low voltage thermostat located in the dining room. This thermostat is wired to control the operation of a solenoid valve placed in the liquid line supplying refrigerant to the cooling coils. When the thermostat calls for cooling, the liquid valve opens and allows refrigerant to flow into the cooling coil. When the vapor pressure in the coil has built up to about 40 lb the low pressure control closes the starting contactor and starts the refrigerating machine, which action automatically starts the circulating fan. When the room temperature has been reduced to that of the thermostat setting, the liquid valve is closed and the machine pumps down until the suction pressure is reduced to about

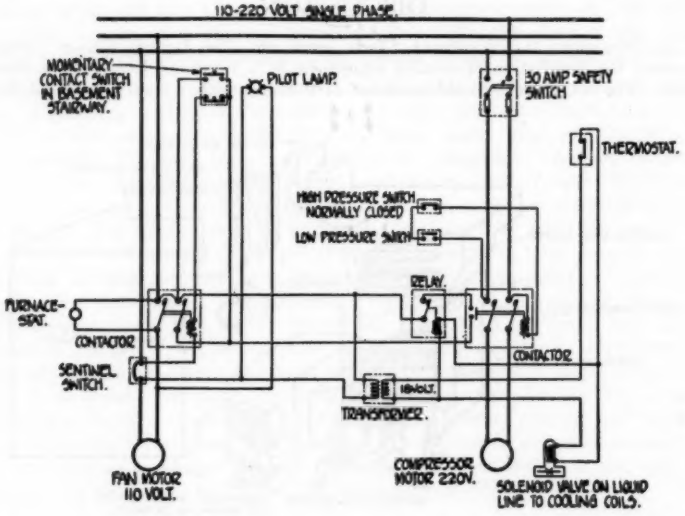

FIG. B. CONTROL DIAGRAM FOR HOUSE COOLING SYSTEM WITH ELECTRIC COMPRESSOR AND DIRECT EXPANSION COOLING COILS

7 lb, at which pressure the low pressure control stops the machine, but the fan continues to circulate air until stopped by a conveniently located manual control. The control diagram is shown in Fig. B.

The system was placed in service on June 12, 1934, and operated for 18 days for a total of 168 hours. Had the system been available on June 1, there would probably have been need for cooling on 5 additional days, according to temperature records. This would make a total of 23 days of artificial cooling for the entire summer. A comparison of the principal operating data with that for the previous two summers, when ice was used, is shown in the summary of operating data on page 233. Two very significant comparisons stand out in this summary; the hours of cooling per day of use are 50 per cent higher than in the previous summer, while the operating cost per hour of cooling is less than one third that for previous summers. The greater number of hours of use can be credited both to the ease with which the

SUMMARY OF OPERATING DATA FOR 1934 SEASON AND COMPARISON WITH SUMMERS OF 1932 AND 1933

	1932	1933	1934
Cooling Medium	Ice	Ice	2-Ton[a] Ref. Machine
Total number of degree hours above 85 F during summer	529	1,408	1,345
Number of days of artificial cooling	22	22	18[b]
Number of hours of artificial cooling	135	134	168
Number of hours of fan use only: Outdoor Air Cooling	57	120	6
Recirculation	...	...	114
Total number of hours of fan use	192	254	288
Average number of cooling hours per day	6.1	6.1	9.3
Electric Energy Used			
1932–33—Ice Water pump and fan—Kwhr	94	142	
1934—Compressor and fan—Kwhr 135 Kwhr for fan and 341 for compressor (Power required by Compressor—2.04 Kw)	....	....	476[b]
Condensing Water for Compressor			
Gallons of water per minute of operation, average	....	....	2.07
Total water consumption for season, gallons	....	....	20,920
Cost of Cooling for Season			
Ice—1932–33	$45.60	$30.60	.....
Electricity at 2¼ cents per Kwhr	2.12	3.20	$10.71
Condensing Water at 20¢/M gal	....	....	4.18
Total operating cost	$47.52	$33.80	$14.89[b]
Average cost per day of artificial cooling	$ 2.16	$ 1.54	$ 0.83
Average cost per hour of artificial cooling	$ 0.35	$ 0.25	$ 0.09

[a] *Condensing Unit:* Refrigerant—dichlorodifluoromethane; 2 cylinder compressor—400 rpm; 2 Hp motor—220 volt—single phase repulsion induction; *Direct Expansion Cooling Coil: Size of Coil Bank*—29 inches × 25 inches × 9¼ inches deep—4.41 sq ft face area; *Face* velocity of air through coils when handling 1,570 cfm—356 fpm; *Actual* static pressure drop through coils 0.11 in. water; *Actual* static pressure drop through entire duct system—0.265 in. water; *Operating* coil pressure—40 to 45 lb per sq in.; *Operating* coil temperature—43 to 48 F.

[b] There were five cooling days in June when machine was not available; corrected cost for the entire season of 23 cooling days = $19.05.

occupants of the home could start the cooling system and to the very low cost of operation. The net cooling accomplished by the machine was approximately the same as that accomplished by the use of ice in the two previous summers.

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TESTS OF COOLING SYSTEM

Test of July 13, 1934

Air Temperatures and Relative Humidity

TIME	OUTDOOR			INDOOR			REGISTER		COOLING COIL	
	W. B. Deg F	D. B. Deg F	R. H. Per cent	Liv. Rm. Deg F	Din. Rm. Deg F	R. H. Per cent	Kitchen Deg F	Din. Rm. Deg F	Entering Air Deg F	Leaving Air Deg F
3:15 P.M.[a]	...	87	..	78	80	62.5	66.5	67	74	65
4:00 P.M.	74.5	90	48	..	..	..	...	...	74	64.5
5:15 P.M.	...	..	..	..	77	53	65.5	66.5	74	64

[a] Cooling started at 2:45 P.M.

CONDENSING WATER

Temperature of water entering condenser—62 F
Temperature of water leaving condenser—83 F
Quantity of water flowing through condenser—2.17 gpm
Heat rejected to cooling water—22,800 Btu per hour

COOLING LOAD

Latent Heat: Moisture condensed from air—2 hour average—3.87 lb per hour
Indoor relative humidity at start of cooling—63%
Indoor relative humidity 2½ hours later—53%
Latent heat load—3.87 × 1056 = 4100 Btu per hour

Sensible Heat:
Air flow = 1570 cfm
1570 × 60 × 0.075 × 0.2375 (74–64) = 16,900 Btu per hour

Total Cooling Load:
16,900 sensible + 4100 latent = 21,000 Btu per hour

Test of July 24-25, 1934

Air Temperatures and Relative Humidity

TIME	OUTDOOR		INDOOR				COOLING COIL	
	D. B. Deg F	R. H. Per cent	Liv. Rm. Deg F	Din. Rm. Deg F	R. H. Per cent	Upstairs S. Bed Rm. Deg F	Entering Air Deg F	Leaving Air Deg F
July 24–3:50 P.M.[b]	100	26	78.5	..	59	..	..	..
July 24–6:00 P.M.	100	..	79	81	..	80	75.5	65.5
July 25–10:00 P.M.	..	..	77.5	..	52	..	..	..

[b] Cooling started at 3:50 P.M. July 24 and continued until 10:00 P.M. on July 25—30-hour run.

COOLING LOAD

Latent Heat:
Moisture condensed from air = 5.06 per hour
5.06 × 1056 = 5,350 Btu per hour

Sensible Heat:
1570 × 60 × 0.075 × 0.2375 (75.5 − 65.5) = 16,800 Btu per hour

Total Cooling Load:
16,800 sensible + 5,350 latent = 22,150 Btu per hour

NOTE: No windows on either floor of the residence were opened during the 30-hour cooling period and no outdoor air was drawn in, the air being completely recirculated.

The performance of the cooling system was measured both by recording instruments and by special tests and the results of two of these tests are given in the accompanying tabulation on page 234.

These test results, together with observations made by the occupants, indicate that the cooling system had sufficient capacity to hold the downstairs temperature to an average value of 78 F and an effective temperature of 71 to 72 deg. On the hottest day during the summer of 1934, July 24, when the outdoor temperature exceeded 100 F for a period of 3 hours, the first floor temperature did not exceed 79 F in the living room. As previous tests had shown that, without cooling, the indoor temperature would have risen to about 85 F under corresponding outdoor conditions, the cooling system can be credited with an average reduction of indoor temperature of 7 F. In addition to this reduction in temperature, the relative humidity was appreciably reduced. Indoor relative humidities at the start of cooling of the order of 67 to 71 per cent were reduced to 60 to 63 per cent at the end of 2 to 3 hours of cooling and the relative humidity at the end of the cooling periods was of the order of 50 to 55 per cent. The amount of moisture condensed from the air by the cooling coils varied between 4 and 6 lb per hour on the days that tests were made.

The performance of the cooling system throughout the summer was quite satisfactory to the occupants of the residence. The most severe test of the system was on the occasion when the compressor was operated continuously for a period of 30 hours, the outdoor temperature exceeding 100 F on the first day and 97 F on the second. Cooling was started at 4:00 p. m. on the first day and continued until 10:00 p. m. on the following day. No windows were opened during this cooling period and, on the second cooling day the upstairs temperature varied between a minimum of 73 F at 6:00 a. m. and a maximum of 79 F at 8:00 p. m. while the downstairs temperature did not exceed 78 F. Although no outdoor air was drawn into the system, the occupants reported that the condition of the air in the sleeping rooms was very satisfactory. The reduction of the indoor relative humidity to a minimum of 52 per cent at the end of the cooling period was undoubtedly a large factor in producing such a satisfactory condition.

As objectionable operating noise can be very detrimental to an otherwise successful cooling installation, the compressor was tested for noise in a laboratory set-up to determine its acceptability for resident use. The compressor is supported at the four corners by laminated cork and rubber pads which in turn rest directly on the basement floor. This very simple machine mounting proved very satisfactory from the standpoint of noise, and, although the flexibility of this mounting permitted an excessive vibration of the compressor as at first designed, this trouble was eliminated later by having the compressor fly-wheel replaced with a specially balanced wheel designed to meet this situation. The noise of the machine was audible in the living room but the occupants considered it in no way objectionable and likened it to the noise caused by a quiet operating oil burner. They did not feel that it was necessary to house the machine in a sound proof box.

The summer's experience with this installation gives rise to certain observations which are deemed of sufficient significance to emphasize in this discussion.

1. In this climate it is quite feasible to cool a moderate-sized residence very satisfactorily with a 2-ton refrigerating machine if the house is well insulated and awnings or blinds are used at the windows. Certainly a 3-ton machine should be large enough even for a house which is only fairly well insulated.

2. Except under unusual circumstances, it is not necessary, nor even desirable, to supply outdoor air to a residential cooling system as natural infiltration will usually

XUM

provide all the ventilation which may be required. Sleeping rooms can be satisfactorily cooled without introducing outdoor air.

3. Since the compressor capacity was somewhat less than would have been required to maintain what may be termed *ideal* conditions, there was no tendency toward overcooling, and the room thermostat could very well have been eliminated and the installation cost correspondingly reduced.

4. There was no objectionable lamp flicker caused by the running of the compressor although the number of fluctuations per second (compression intake cycle) was 9.7, which is rather low. The most troublesome frequency has been found to be about 6 or 7 fluctuations per second. Laboratory tests were made on three well known makes of compressors of 2 to 3 ton capacity, with fluctuation rates ranging from 9.7 to 39 per second, and it was found that no really noticeable light flicker would result under conditions usually found in residential lighting circuits.

5. A total operating cost of $19.00 for a cooling season can certainly be considered very moderate, and should be no obstacle to the growth of this class of comfort cooling.

With regard to the section of Professor Kratz' paper which deals with the subject of cooling with outdoor air at night, the writer would like to point out one factor which is not stressed in his paper. Fig. 5 shows the influence of the rate of circulation of night air on the cooling effect in the Research Residence by plotting the number of air changes per hour against the slope of the inside temperature drop curve. While this curve very nicely brings out the relationship between rate of air change and the effectiveness of this air movement in reducing inside temperature, there is danger that the reader may get the impression that this is a true measure of the effectiveness of cooling. In addition to cooling effect caused by the inside temperature drop, there should be considered the additional cooling effect produced by the movement of air. Our experience with attic ventilating fan installations indicates that this effect is almost as important in producing comfort as is the reduction in inside temperature. Some home owners who had attic ventilating fans installed this past season have particularly commented to the effect that it was the movement of air which they felt was most conducive to comfort. In other words, curves showing the drop in indoor temperature produced by attic ventilating fans do not tell the whole story of the improvement in indoor air conditions, and it is well to stress the fact that air movement caused by attic fans contributes a great deal toward producing the more comfortable conditions noted by the owners of these installations.

E. A. JONES: What was the temperature carried in the cooling coil? What was the minimum temperature of air introduction, and was any condensation noted on the distribution ducts? Was the air distribution in the house such that ideal conditions, as far as circulation is concerned, were maintained?

ALBERT BUENGER: I should like to ask Professor Kratz whether, in the curve showing the loads, if he has used a planimeter or simply arrived at the total cooling load of the actual compared with the computed over the entire 24 hours operation.

W. L. FLEISHER: I would like to know whether the authors made calculations or took note of the number of people in the enclosure, the variation in their delivered temperature with changes in the outside wet-bulb, and the calculations of the amount of infiltration into the house and its relationship to the number of people in the house, which could probably be calculated from the amount of condensate taken out in the coils. Was anything of that kind done?

A. P. KRATZ: In response to Mr. Helmrich's paper, it is very gratifying to learn that their results check our results so closely when the two are reduced to the same basis.

In answer to Mr. Jones' question relative to the temperature in the cooling coil, the temperature of this coil was measured at three points: at the entering point it was 42 F; at the control point, which was about two-thirds of the way through the coil, it was about 57 F, as I recollect it; and the leaving temperature was 75 F. The temperature of the air at the registers was approximately 70 F, and there was a rise of about 4 F from the air leaving the coil to the air leaving the registers; so it would seem that the air leaving the coil was approximately 66 F, and at the registers, about 70 F.

We had no condensation on any of the ducts.

MR. JONES: May I ask what the wet-bulb temperature was leaving the coil?

PROFESSOR KRATZ: The wet-bulb temperature leaving the coil was approximately 60 F.

In answer to Mr. Jones' third question about the circulation, in common with the two previous years' tests, we found no difficulty with distribution of the air throughout the house. We had no appreciable drafts nor objectionable drafts, and as far as we were able to determine the air was quite uniformly distributed through the house.

In answer to Mr. Buenger's question, we did not try planimetering the area of the curves to compare the total load under the calculated curve with the total load under the actual curve, but that could easily be done.

In reply to Mr. Fleisher's question, our load was calculated on the basis of occupancy of between 3 and 4 people in the house. If I understand correctly, you mean that in calculating the load, did we take account of the wet-bulb temperature outdoors as well as the dry-bulb temperature in obtaining the theoretical or the calculated moisture load? We did.

MR. FLEISHER: I wanted to see whether you were measuring your condensate and if from that you had been able to determine the amount of outdoor air brought in from the delivered air, and the outside wet-bulb of the air or dew point; my thought being that if you had a variation in your outside absolute humidity, you then necessarily would either have had to put on additional refrigeration to maintain the same condition, or you would have changed the conditions indoors. Was any record of that kept, or were you interested almost entirely in dry-bulb conditions, letting the relative humidity take care of itself?

PROFESSOR KRATZ: We had to let the relative humidity take care of itself, since we controlled entirely on the dry-bulb temperature. We did weigh the condensation, and it checked closely with the amount of condensation that we should have obtained from about one air change per hour.

MR. FLEISHER: My whole reason for asking that question was that under average conditions the amount of heat that was removed in the way of latent heat being, as I recall the figures, 25 per cent of the total, is less than the usual amount if the proper amount of outside air is introduced.

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PROFESSOR KRATZ: We never could get an exact check on those values, but it checked roughly with the amount of infiltration for one air change per hour. In the previous summer's work in which we did not introduce any air from the outside, we obtained an infiltration loss of approximately three-quarters of an air change per hour. This summer the same comparison seemed to indicate that our one air change per hour that we were taking in from the outside was about all the infiltration that occurred.

No. 1013

SEMI-ANNUAL MEETING, 1935

THE members of the Society responded with enthusiasm to the invitation of Ontario Chapter and a fine three-day meeting was enjoyed at the Royal York Hotel, Toronto, June 17-19. Members came from the Atlantic to the Pacific and many made this trip the occasion for a vacation tour, visiting Ottawa, Montreal, Quebec, Thousand Islands, Sault Ste. Marie, and other points of historic interest.

On each of the three days in Toronto many sessions were held to discuss technical papers and reports. Attendance at the meetings was exceptionally heavy, indicating that the Program Committee had selected subjects that were of unusual interest.

Pres. John Howatt called the 1935 Semi-Annual Meeting of the Society to order in the Ballroom of the Royal York Hotel, Toronto, and M. W. Shears, President of the Ontario Chapter, made a brief speech of greeting.

President Howatt expressed the appreciation of the members for the welcome given on behalf of the Ontario members.

Amendments to By-Laws

The first item of business was consideration of two amendments to the By-Laws which were prepared by the Committee on Constitution and By-Laws and approved by the Council and submitted to all members, in accordance with Article B-XVI as follows:

Add the following new section to Article B-VIII:

Section 11. At the first meeting of the Council after the Annual Meeting, or as soon thereafter as possible, the president shall appoint a Committee of at least five (5) members, whose duty it shall be to arrange for the publication of THE GUIDE. This Committee shall be responsible for the text section and shall arrange for the inclusion of such new chapters as will furnish information regarding new developments.

In order that the wording of the various Sections of Article B-VIII may be in agreement, the Council suggested a change in the wording of the present *Section* 11, which would become *Section* 12.

Section 11. At the organization meeting of the Council, each year, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This Committee shall function according to the terms of the F. Paul Anderson Award. (See Appendix B.)

To Be Amended as Follows: Section 12. At the first meeting of the Council after

the Annual Meeting or as soon thereafter as possible, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This Committee shall function according to the terms of the F. Paul Anderson Award. (See Appendix B.)

It was moved and seconded that the new *Section* 11 of Article B-VIII and the amended *Section* 12 be adopted. This motion was unanimously carried, and President Howatt announced that these new sections would become effective immediately.

President Howatt had a special word of greeting for three of the Society's Life Members who were in attendance, J. D. Cassell, Philadelphia, J. M. Stannard, Chicago, and Warren Webster, Camden, N. J.

Just before adjournment, announcement of the voting on the Garage Ventilation Code was given. A tabulation made by the Board of Tellers, Messrs. H. W. Fiedler, W. M. Heebner and H. B. Eells, gave the following results: *for* the Code 215, *against* the Code 5, *total legal votes* 220, *total votes cast* 222.

The Second Session of the Semi-Annual Meeting was called to order by President Howatt on Tuesday morning and the technical papers on the program were presented and discussed.

The Wednesday morning session was opened by Pres. John Howatt at 9:30 a.m.

Proposed Amendments

Under the heading of new business, F. C. McIntosh submitted two amendments to the Constitution, relating to Student Membership, and explained that the lower age limit suggested would be helpful to those technical schools and colleges which were extending their courses in heating, ventilating and air conditioning. The following amendments were presented on petition of 19 members.

ART. C-II, *Section* 7. A Student Member shall be a person between the ages of 20 and 25 years, who is regularly attending courses in an engineering college or technical school at the time of applying for membership.

To Be Amended as Follows: A Student Member shall be a person between the ages of 18 and 25 years, who is regularly attending courses in an engineering college or technical school at the time of applying for membership.

ART. C-III, *Section* 4. When a Student Member discontinues his regular studies in an engineering college or technical school, it shall be incumbent upon him to apply within one (1) year for advancement to either Junior or Associate grade. In no case shall the period of Student Membership exceed four (4) years.

To Be Amended as Follows: When a Student Member discontinues his regular studies in an engineering college or technical school he shall be automatically transferred to Junior grade of membership unless he has applied for and has been elected to Associate grade. In no case shall the period of Student Membership exceed four (4) years.

On motion, properly seconded, it was

Voted: To refer this matter to the Committee on Constitution and By-Laws for presentation to all members of this Society in the prescribed manner.

Tribute to Stewart A. Jellett

C. V. Haynes, chairman of a special committee appointed by President Howatt, offered a resolution on the late Stewart A. Jellett who was a Charter Member, an Honorary Member and the first man to serve a full term as President of the Society.

Whereas, Stewart A. Jellett was a Charter Member and the first man to serve a full term as President of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and

Whereas, the Society has honored him by conferring Honorary, Life and Presidential Membership upon him as a recognized leader in the development and advancement of heating and ventilating, and

Whereas, he was a pioneer in the profession and proved his ability as an engineer and executive, and

Whereas, he also served the Society as secretary on committees and in an advisory capacity during its 41 years of existence, and

Whereas, it has been the will of Almighty God to take him unto Himself, therefore

Be It Resolved, that we, the members present at the Semi-Annual Meeting 1935, in Toronto, Canada, hereby express our deepest regret at his passing but rejoice in having been associated with one of nature's noblemen who endeared himself to all of our members, and

Be It Further Resolved, that this resolution be inscribed upon the proceedings of the Society and a copy be presented to the bereaved family so that they may know with what high esteem his memory is held.

C. V. HAYNES, *Chairman.*
H. P. GANT,
J. F. HALE.

On motion, properly seconded, this resolution was unanimously adopted.

Resolutions

O. J. Prentice, Chicago, presented formal resolutions of thanks and appreciation for the splendid handling of the meeting and the enjoyable program that had been provided.

The Committee on Resolutions has a particularly happy privilege. This privilege, however, even at technical sessions, presents a problem. The Committee feels that suitable appreciation of the wise planning, the untiring energy and the self-sacrificing devotion of those who have made this meeting so outstanding a success cannot be expressed in a formal—"Be it resolved—whereas, and whereas, and whereas," but that such expression should emulate the friendliness which has characterized these stimulating three days.

Be It Resolved, that the thanks and appreciation of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in Convention assembled be expressed—

1. *To the Ontario Chapter,* for a hospitality superlative alike in its graciousness, in its completeness and in its never failing anticipation of our every comfort and need. A hospitality that has been the sum total of the individual contribution of each member of the Chapter and which finds its completeness in the never-to-be-forgotten friendly and efficient contribution by the ladies.

2. *To the City of Toronto,* which through its officials and home loving citizens

has made us feel at home and which, through the whole hearted cooperation of the Police Department in transportation matters and in the unfailing courtesy of the policemen on the streets, has made our sojourn a happy experience in a city of brotherly love.

3. *To the Convention Bureau and the Local Press,* for the publicity given to our Society and to the Sessions of this Semi-Annual Meeting in particular.

4. *To the Ontario Research Foundation, the Harbor Commission and the Various Clubs and Organizations,* which have so generously added to the enjoyment of our visit.

5. *To the Authors of Papers,* for their investment of time in preparation and for their presentation of important subjects touching the welfare of humanity.

6. *To the Trade Press,* for its advance publicity of this Meeting, and for its constant interest in the work of the Society and publication of its doings.

7. *To the Royal York Hotel and its staff,* for a practical interpretation of the friendliness of the city, and of the Ontario Chapter, an interpretation which has made our residence here most comfortable.

O. J. PRENTICE, *Chairman.*
R. V. SAWHILL,
W. E. STARK.

On motion, properly seconded, these resolutions were unanimously adopted. The meeting adjourned.

PROGRAM SEMI-ANNUAL MEETING, 1935

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
ROYAL YORK HOTEL, TORONTO, ONT., CAN.

JUNE 16-19, 1935

(All events scheduled on Daylight Saving Time)

Sunday, June 16, 1935

6:30 P.M. Dinner and Meeting of the Council

Monday, June 17, 1935

9:00 A.M. Registration

10:00 A.M. Greetings by M. W. Shears, President, Ontario Chapter
Response by President Howatt
Amendments to By-laws
Technical Papers:
Characteristics of Registers and Grilles, by J. H. Van Alsburg
A Study of Classroom Ventilation:
Part I—Classroom Odors with Reduced Outside Air Supply, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and M. F. Lichtenfels
Part II—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels
Part III—Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and M. F. Lichtenfels

1:00 P.M. Cars leave hotel for Weston Golf Club

1:30 P.M. Meeting of the F. Paul Anderson Award Committee, G. L. Larson, Chairman

1:30 P.M. Meeting of the Committee on Research, A. P. Kratz, Chairman

1:30 P.M. 18-hole Handicap Golf Tournament for Research Cup

3:00 P.M. Sight-seeing trip for ladies through City of Toronto and visit to Granite Club, returning to the hotel about 5 P.M.

6:30 P.M. Buses leave hotel for the Old Mill to join the golfers

7:30 P.M. Informal dinner followed by dancing for members and ladies at the Old Mill

Tuesday, June 18, 1935

9:30 A.M. *Technical Papers:*
Heat Absorbing Glass Windows, by W. W. Shaver
Cooling and Air Conditioning in an Ale Brewery, by C. P. Creighton and F. J. Friedman
An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan
A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, F. O. Urban and L. Levine

10:30 A.M. Inspection of Chinese Porcelain Rooms at the Ontario Museum by the ladies

12:30 P.M. Luncheon Meeting of the Committee on Ventilation Standards, W. H. Driscoll, Chairman

12:30 P.M. Get-Together Luncheon Chapter Officers

1:30 P.M. Meeting of the Guide Publication Committee, G. L. Larson, Chairman

1:30 P.M. Informal golf matches for members

1:45 P.M. Trip to Royal Canadian Yacht Club, where the ladies will enjoy a Bridge and Tea. Cars leave hotel at 1:45 P.M., returning about 5:00 P.M.

2:30 P.M. Visit and Inspection of Ontario Research Foundation

2:30 P.M. Boat Trip and Inspection of Toronto Harbor Development for men

7:30 P.M. Annual Banquet and Dance for members and ladies at the Royal York Hotel

Wednesday, June 19, 1935

9:30 A.M. *Technical Papers:*
Oil Burning in Residences, by D. W. Nelson
The Dust Problem in Air Conditioning, by F. B. Rowley
Chicago Standards for Certified Air Conditioning, by S. R. Lewis
Resolutions
Adjournment

10:00-12:00 A.M. Shopping tour of Toronto stores for ladies

12:30 P.M. Luncheon meeting of the Nominating Committee

COMMITTEE ON ARRANGEMENTS

E. H. Gurney, *General Chairman*
L. L. Anthes, *Vice-Chairman*
H. H. Angus, *Reception*
Mrs. E. H. Gurney, *Ladies*
H. B. Jenney, *Ladies*
W. P. Boddington, *Entertainment*
Thomas McDonald, *Banquet*
E. R. Gauley, *Publicity*
M. F. Thomas, *Finance*
J. S. Paterson, *Registration*
M. W. Shears, *Transportation*
W. R. Blackhall, *Past President*, Ontario Chapter
H. R. Roth, *Secretary*, Ontario Chapter

No. 1014

CHARACTERISTICS OF REGISTERS AND GRILLES

By J. H. VAN ALSBURG* (*MEMBER*)
HOLLAND, MICH.

DURING the last few years when rapid progress has been made in the air conditioning industry, emphasis has been placed upon those phases of the installation which seemed to be most important with the result that little attention was paid to the effect which the registers or grilles have on the performance of the system. Now it is apparent that registers and grilles directly affect the system in the following respects: 1. Noise, 2. Resistance, 3. Volume, 4. Flow, 5. Appearance.

An attempt will be made in this paper to outline the latest information available regarding the effect of registers and grilles on these five characteristics. This information is based on the results of many tests carried on since 1922. Several types of grilles are discussed, but no attempt to favor any particular manufacturer's design is intended, nor is any pretense made that the information given is final and complete. Much remains to be done on this subject and it is hoped that this paper will serve to stimulate further research and investigation.

At this point it is desirable to define the terms used in this paper. A grille is primarily a grating, used to conceal or decorate an opening through which air passes. A grille has no provision for controlling air volume. A register is the combination of a grille and some means for regulating air flow. A supply grille is one which admits air *into* a room or other enclosed space. A recirculating grille is one through which air is withdrawn from the room. Unless otherwise stated, the information given in this paper applies to supply grilles.

TEST APPARATUS

A room approximately 40 x 20 x 12 ft was used as a test laboratory. The testing apparatus, as shown in Fig. 1 consists of a standard blower, discharging a volume of air into a horizontal plenum chamber. From this chamber the air passes into a vertical duct, which makes a right angle expanding approach to the grille to be tested.

These tests were made with typical grilles of standard manufacture, all size 12 x 6, as shown in Fig. 2.

* Engr., Hart & Cooley Mfg. Co.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June, 1935.

Smoke bombs were used to determine the path of the air stream and a standard 4 in. anemometer, which was recently calibrated, was used for all velocity readings. In making the velocity readings at the register to determine the average velocity, the grille was blocked off in squares with chalk and one minute readings were taken of each block to determine an average for the total area.

In checking the velocity of the air away from the grille to determine the path of the air stream, the anemometer was used as described under Methods of Testing. A pitot tube was used with a special manometer for making all pressure readings, which were taken in the vertical riser above the plenum chamber, midway between the plenum chamber and the approach. These readings were taken by making a traverse to establish an average for the pressure at this point.

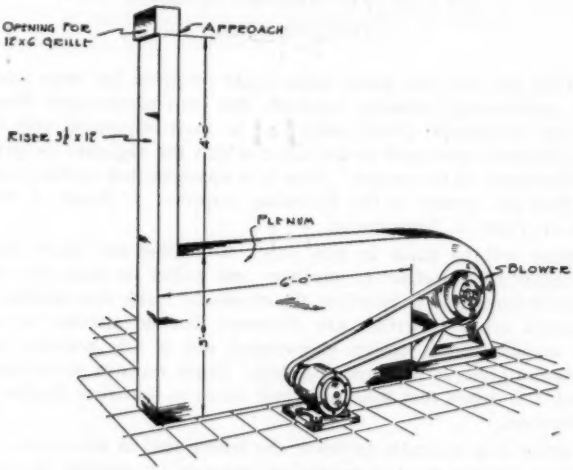


FIG. 1. TEST SET-UP

No suitable test apparatus was available for determining the noise level, and the blower, being in the test room near the grille, made it difficult to determine the cause of the noise. Therefore in the absence of suitable noise measuring devices, noise was determined by the hearing of several individuals assisting in this test.

METHOD OF TESTING

Lines were drawn on the test room floor as shown by Fig. 3, to serve as guides in taking velocity readings. Readings were taken at all intersections of these lines at 4 in. intervals above and below the center line of the grille at as high or as low a level as required to record the entire stream. No attempt was made to record velocities below 50 fpm.

Except where otherwise specified on the figures shown, the air temperature

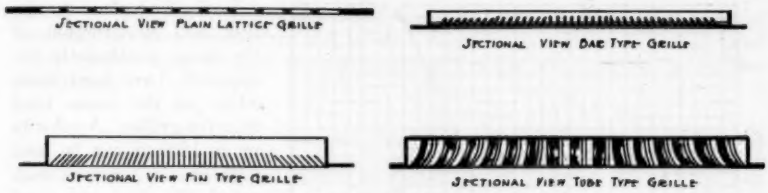

JECTIONAL VIEW PLAIN LATTICE GRILLE
JECTIONAL VIEW BAR TYPE GRILLE
JECTIONAL VIEW FIN TYPE GRILLE
JECTIONAL VIEW TUBE TYPE GRILLE

FIG. 2. GRILLES TESTED

is room temperature of 70 deg. All grille areas were based on the Davies Formula, which is as follows for registers:

$$\text{Cfm} = CV\frac{A+a}{2}$$

and for this equation CV is a correction factor to be used for various velocities and anemometers of different sizes, A is the gross area of the grille and a is the net area of the opening through the grille, which is divided by 2 to establish the average or effective area. This formula has been thoroughly covered by papers [1] presented before the Society on three different occasions.

NOISE

As mentioned before, suitable apparatus for accurately determining the cause of noise and its level was not available. The noise level for grilles has been considered 35 decibels but this does not take into consideration the pitch or frequency of the noise and this suggests further research be conducted to

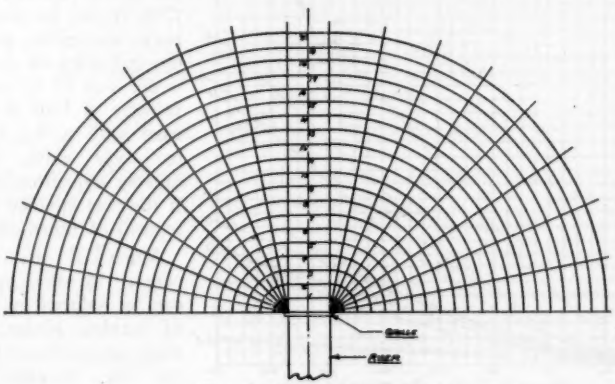


FIG. 3. FLOOR LAYOUT

determine noise and that improvements be made in the present noise level determination equipment. Even without suitable apparatus certain results were obvious from the tests made and are herein mentioned.

[1] Measurement of the Flow of Air Through Registers and Grilles, L. E. Davies, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930; Vol. 37, 1931; Vol. 39, 1933.

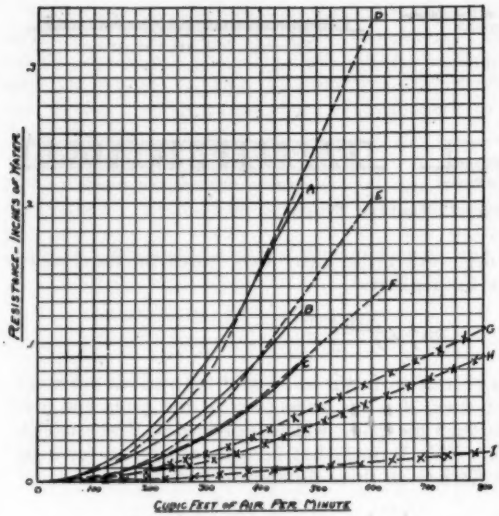


FIG. 4. GRILLE RESISTANCE

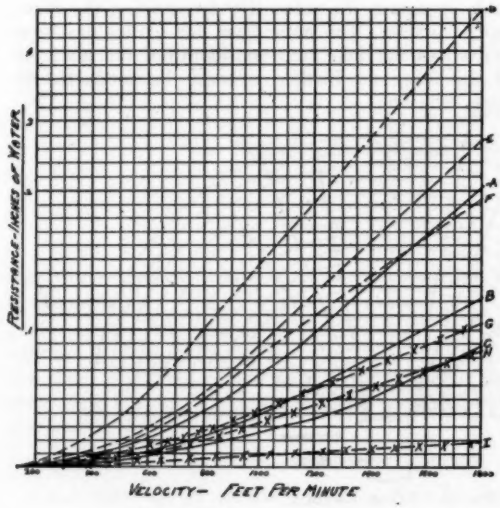


FIG. 5. GRILLE RESISTANCE

SYMBOLS USED—FIGS. 4 AND 5

A—Perforated grille and duct
B—Perforated grille only
C—Duct only
D—Fin, bar or tube grilles and duct
E—Fin, bar or tube grilles only
F—Duct only
G—Tube grille, turning blades and duct
H—Tube grille and turning blades
I—Turning blades and duct

In the first place the design and construction of the ducts, particularly the approach, have much more effect on the noise level than the grilles. A velocity up to 1500 ft can be used without air noise with practically any type of grille, using a standard approach, and by standard approach reference is made to a right angle turn between the duct and the grille with a change in dimension so the perimeter of the grille flange is larger than the perimeter of the duct, commonly called a riser, to allow slower outlet velocity. It is very difficult to eliminate noise in an air conditioning system with a duct velocity of over 2000 ft, which would be necessary if a velocity of 1500 ft were recorded at the grille. A velocity of 1700 ft can be used with perforated grilles, provided the fret work of the grille is not over ¼ in. wide. A velocity of 1500 ft can be used with a fin, tube or bar type grille without causing objectionable noise if the rear surface is flush instead of indented.

Much of the noise caused by the approach can be reduced by the use of turning blades, which allow an increase of 300 ft for the velocities mentioned, without increasing the noise caused by the

grille. A properly designed combined approach, turning blades and grille in one unit, has been constructed that permits velocities as high as 3500 ft without objectionable noise because the air turbulence in the approach and grille was eliminated.

RESISTANCE

Figs. 4 and 5 show the resistance of various types of grilles, and of the duct only, based on cubic feet of air per minute and velocity in feet per minute, respectively.

All grilles used for the tests were 12 x 6 in., the construction of the grilles being shown in Fig. 2.

The Figs. 4 and 5 indicate that the resistance caused by the grilles will never

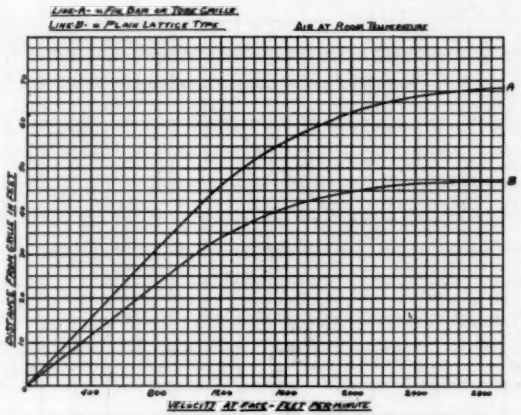


FIG. 6. LINEAR CARRY OF AIR STREAM

exceed 10 per cent of the total resistance of the system, and that therefore grille resistance is of relatively small importance.

VOLUME

A change in the net free area of a grille does not have a corresponding effect on volume. For example, a 10 per cent reduction in the net area will reduce the volume approximately only 5 per cent provided the net area of the grille remains over 50 per cent.

AIR FLOW

By air flow is meant the distribution of incoming air into the room. This involves the distance the air travels, and the direction and shape of the air stream. It is obvious that the correct distribution of the incoming air is an important factor in the comfort of the occupants of the room and it is perhaps in connection with air flow that grilles assume the greatest importance. It is

also in connection with air flow that the performance of different types of grilles varies the most.

There is no one type of grille which is suitable for all conditions. Much depends on the location of the grille, size of the room, velocities used, temperature of the incoming air and other factors. The characteristics of various types of grilles will be explained in order to assist in selecting the proper type to be used under whatever conditions exist on the installation.

The length of the air stream is governed primarily by the velocity of the air;

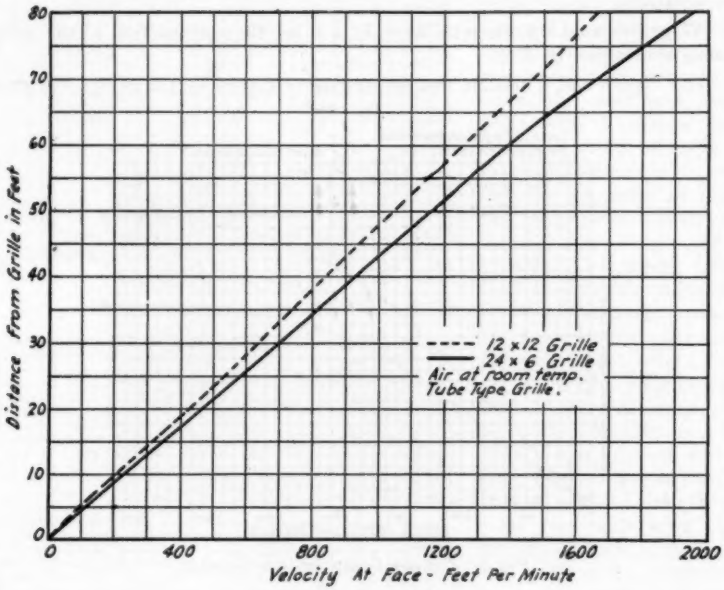


FIG. 7. LINEAR CARRY OF AIR STREAM

secondly, by the type of grille; and thirdly, by the proportions of the grille. Fig. 6 shows the air carry for two types of grilles of the same size at various velocities.

Fig. 7 shows the air carry for two grilles of the same type and area but of different proportions.

The results shown in Fig. 7 indicate that the air will travel further with a grille having a smaller perimeter than one having a large perimeter, other conditions being the same.

The direction and shape of the air stream is very materially affected by the type of grille and also by the nature of the approach. If it is desired to project the air on a horizontal plane, it is necessary to either use turning blades in the approach or else use a grille having horizontal bars or fins of sufficient depth to guide the air horizontally. The air can be readily directed to the left or

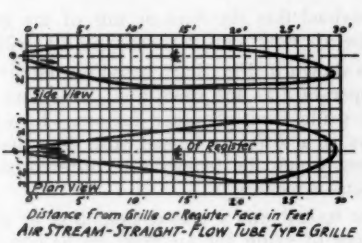
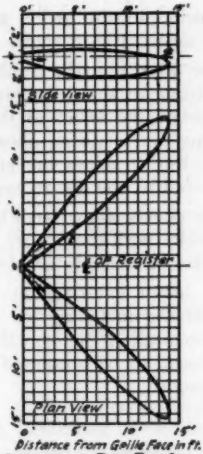
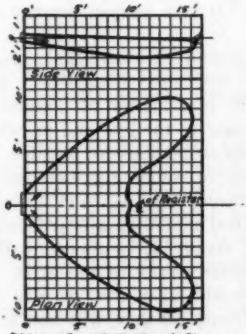
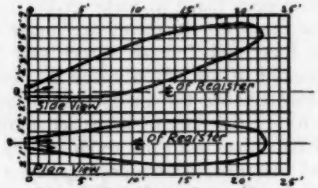


FIG. 8. AIR FLOW AT ROOM TEMPERATURES




Side View
Plan View
Distance from Grille Face in Ft.
AIR STREAM-WITH TUBE-TYPE GRILLE
2° AT 45° LEFT-3° AT 45° LEFT TO 5° LEFT
1° STRAIGHT-3° AT 45° RIGHT TO 5° RIGHT-2° AT 45° RIGHT.



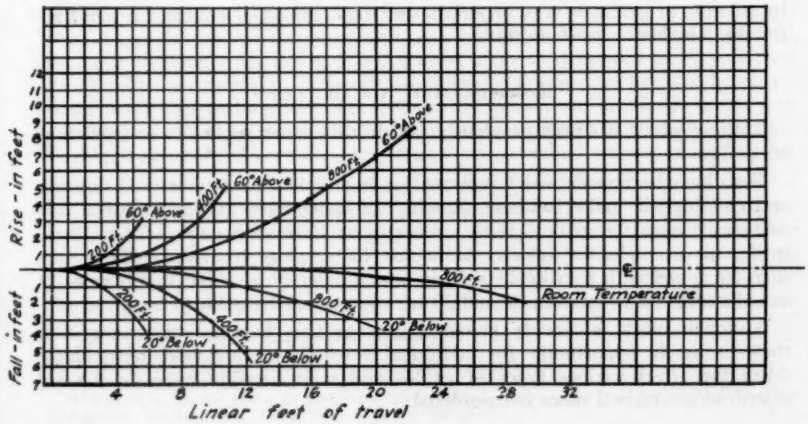


FIG. 9. AIR STREAM RISE AND FALL AT VARIOUS VELOCITIES AND TEMPERATURES

right by means of the grille, provided that the bars or fins of the grille are sufficiently deep. Fig. 8 shows the air flow with various types of grilles.

It will be noted that the results shown in Fig. 8 are based on the air leaving the grille being at the same temperature as the room air. This was done in order to isolate the effect of the grille from all other factors. In actual use the incoming air is almost always at a different temperature than that of the room air. Fig. 9 shows the effect which the temperature differential has on the rise or fall of the air at various velocities.

From the charts shown on Figs. 6, 7, 8, and 9 it is obvious that by using the proper velocity with the proper type of grille, the air flow can be controlled as desired with considerable accuracy.

Another factor entering into the distribution of the air in a room is the effect which the incoming air has on the air already in the room. This is called the aspiration factor. Information regarding this has not been sufficiently developed to be presented at this time. It is a subject worthy of much additional study.

APPEARANCE

What constitutes an attractive looking grille is a matter of opinion. Many attempts have been made to make grilles pleasing to the eye and in other cases attempts have been made to make grilles as inconspicuous as possible. At the present time many developments are under way to combine grilles and lighting fixtures so they will furnish direct and indirect lighting and to carry the required amount of air into a space, so designed that they are artistically proportioned and enhance the appearance of any room. It is important that it be impossible to see the duct or approach immediately back of the grille and that the finish harmonizes with the room. They should, of course, be neat and trim looking and installed, where possible, in a frame so they will not streak the walls, which can be accomplished by setting the grille in a frame so the front of the grille is flush with the wall, which causes the air that leaks between the grille and the frame to travel at right angles to the surface of the wall so it will not streak as it often does when the air travels parallel to the wall surface. Installation of grilles is very important and the architect or engineer responsible for the job should insist on this.

RECIRCULATING AIR GRILLES

Nothing which has been previously said in this paper applies to recirculating air grilles.

Tests have been made which indicate that high velocities may be used through recirculating air grilles without showing an appreciable air movement a few feet away from the grille. With a velocity of 1000 ft through a 12 x 12 in. grille, for example, the velocity of the air 16 in. away from the grille is less than 50 ft and with a grille 24 x 24 in. and a velocity of 1000 ft, this velocity is reduced to 50 ft between 24 and 30 in. away from the grille.

Although much is already known regarding the characteristics of grilles, there is ample opportunity for additional research in this field, particularly when the effect various types of grilles and their location have on the air distribution within a room is considered.

No. 1015

CLASSROOM ODORS WITH REDUCED OUTSIDE AIR SUPPLY

By F. C. HOUGHTEN * (*MEMBER*), H. H. TRIMBLE,** CARL GUTBERLET,*** AND MERLE F. LICHTENFELS † (*NON-MEMBERS*)
PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

AIR volume requirements for school house ventilation probably have been the subject of more controversy than any other phase of heating and ventilation. The engineer, the medical man, the educator, and the school business management have all had their part in the developments. The wide diversity of opinion more or less accepted by various groups at different times is indicated by the lack of unanimity in the legal requirements on the statute books of several States. These requirements vary from 30 cu ft of outside air per minute per pupil, downward.

At times, the engineer and members of the medical profession have seemed to be far apart in their positions on this controversy. The engineer approaching the subject from the practical demands of school ventilation requirements knew that he could not maintain acceptable conditions within the school room without handling a large volume of air through the prevailing ventilating system in the classroom of conventional size and pupil density. Thus convinced, he naturally sought for and accepted diverse theoretical reasons to support what he found to be necessary. The representative of the medical profession, on the other hand, approaching the subject from his theoretical knowledge concerning the oxygen requirements and the lack of harmfulness of carbon dioxide, was unable to recognize a need for more than a very nominal amount of outside air.

The past decade and a half have witnessed the development through research of a more complete understanding of the processes of metabolism and thermal exchange between the human body and its atmospheric environment, including the effect of the temperature, humidity and motion of the air on a person's feeling of warmth and the necessity for a reasonably high air movement within the occupied space (as measured by the Kata thermometer) in order to insure

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the distribution of air having the proper physical qualities over the body itself. These developments have resulted in complete vindication of the engineer's position regarding the circulation of a relatively large volume of air in the conventional classroom. It has developed, however, that the recirculation of air within the classroom is as effective in providing the required motion and cooling power of the air as the supplying of outside air, provided the proper effective temperature of the recirculated air is maintained.

This later development has reopened the controversy concerning the required amount of outside air. If recirculated air is as effective as outside air in producing the desired cooling power of the air, why should an outside air supply greater than that necessary to supply the minimum requirements for oxygen supply and carbon dioxide elimination be required? Although recently challenged by some authorities, it seems to be generally agreed that the requirement of outside air for oxygen supply and carbon dioxide elimination is no greater than from 2 to 4 cfm per person, which in conventional classrooms will usually be supplied by infiltration and the initial volume of air in the room before occupancy. Based upon this reasoning, it seems logical to contend that no outside air need be supplied through the ventilating system, except when required to maintain the desired cooling power of the air. Recent research aimed to establish a more rational basis for outside air requirements in school ventilation has indicated that when outside air is not required to maintain the cooling power, odors represent a limiting factor. Lehmberg and others[1] have recently shown that in a small confined space, odors become objectionable when the air supply drops to from 10 to 25 cfm per person.

The Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has recently been engaged in a study of the relation of air delivery in a classroom to drafts. In connection with this study, facilities were found available for obtaining first-hand practical information on the relation between air delivery to a classroom and such effects as overheating, change in moisture content, increase in CO_2 content, and prevalence of odors.

The study was made in classrooms of the Woodlawn Avenue School of Munhall, Pennsylvania, a steel mill suburb of Pittsburgh. The Woodlawn Avenue School includes pupils of the 7th, 8th, 9th and 10th grades, from the better class residential sections of the town, farthest removed from the mill district.

The results presented in this paper are based on three half-day studies and several additional observations made in classrooms normally occupied by classes. Study Number One was made in a first-floor classroom on a day when the outside temperature ranged from 10 F at 8:00 a.m., to 14 F at noon. A cold day was chosen with the expectation that overheating would not result from reduction in outside air supply. The particular room studied was chosen for the reason that, because of the prevalence of cold drafts, the teacher was unusually amiable toward the study, even though the observations made might be somewhat disturbing to her work.

This room had a 28 ft eastern exposure containing approximately 36 per cent glass. It was 22 ft wide and had a 12 ft ceiling. The lower half of the windows opened on a central horizontal pivot. The movable sections were weather-

[1] A Laboratory Study of Minimum Ventilation Requirements: Ventilation Box Experiments, by W. H. Lehmberg, A. D. Brandt, and Kenneth Morse, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935, p. 157.

stripped and locked firmly in place so as to allow little opportunity for air leakage when closed. The room was equipped with a unit ventilator controlled from the janitor's quarters in the basement either to take in all outside air or to recirculate room air. This unit was equipped with a heating unit supplied with steam without either automatic or convenient hand control. A thermostatic controlled damper directed the air either through the heater or around it. The room was also equipped with one auxiliary cast-iron direct radiator located along the exposed wall, and supplied with steam under thermostatic control. The control of the by-pass damper in the unit ventilator and of the steam supply to the auxiliary radiators was had by a dual thermostat designed first to cut off

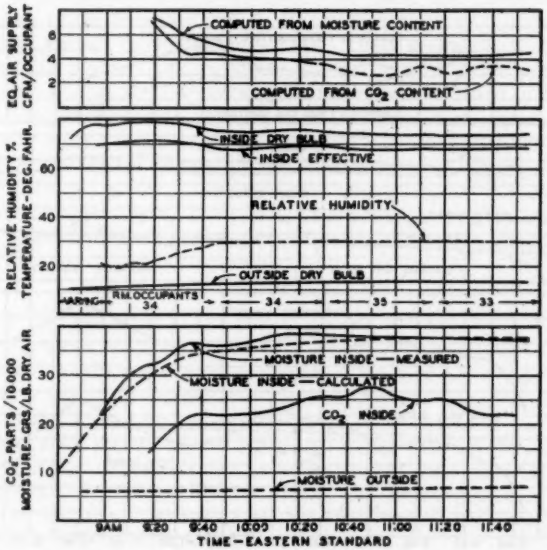

FIG. 1. RELATION BETWEEN AIR CHANGE IN A CLASSROOM AND OCCUPANCY. CO_2 PRODUCTION AND INCREASE IN MOISTURE CONTENT. NO POSITIVE AIR SUPPLY. OVERHEATING FROM SOLAR RADIATION AND BOILER PLANT

the steam supply to the direct radiator as the temperature rose through the desired point, and then if this did not arrest the rise in temperature, to open the by-pass damper in the unit ventilator. In this room the unit ventilator was located under a window in the center of the front third of the room, or near the teacher's desk. The vent stack was 20 in. by 30 in. in cross-section, with a 20 in. by 38 in. opening immediately above the floor line in the rear of the room.

Before the arrival of the children on the morning of the study, changes were made so that the steam supply to the auxiliary radiator could be controlled by hand. The central control of the outside air intake and of the recirculating dampers was disconnected, the intake damper was fastened securely in the

closed position, and the recirculating damper in the open position, and the steam supply was shut off from the heater in the unit ventilator. The vent flue opening was closed.

The unit ventilator was allowed to operate at the usual rate of air circulation, and an attempt was made to maintain a room temperature of 70 F by hand control of the steam to the auxiliary radiator, beginning at 8:30 a.m., or one-half hour before the beginning of the first class period.

A Haldane gas analyzing machine was set up in the center of the room, with which samples of the air were taken and analyzed continuously from 8:30 a.m. until 12:00 noon. Another observer recorded the dry-bulb and wet-bulb

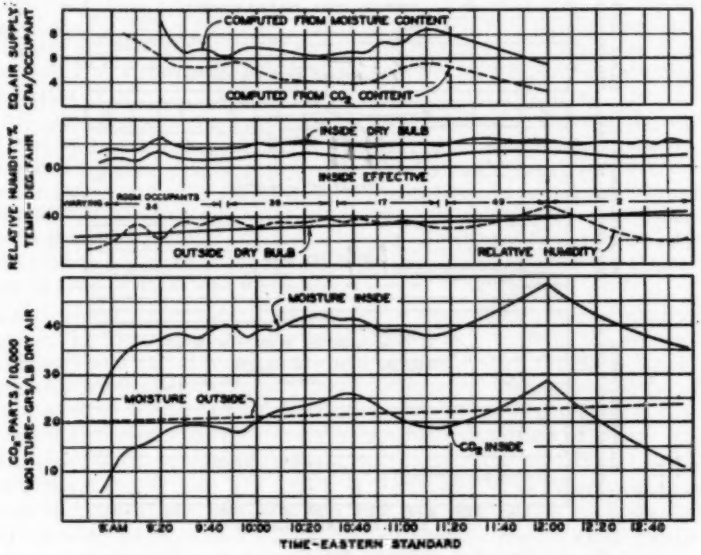

FIG. 2. RELATION BETWEEN AIR CHANGE IN A CLASSROOM AND OCCUPANCY. CO_2 PRODUCTION AND INCREASE IN MOISTURE CONTENT. NO POSITIVE AIR SUPPLY. NO SOLAR RADIATION

temperatures and other conditions in the room, besides making the desired adjustments to the heat supply. There was a 6 to 10 mph, NW wind during this study. The occupants of the room had an average body surface area of 15.2 sq ft. The results of this study are plotted in Fig. 1.

The second study was made in a room on the opposite side and on the third or top floor of the building, having an exposed wall 32 ft long, containing approximately 35 per cent glass. This room was 22 ft wide, and was also heated by a unit ventilator and two auxiliary direct, cast-iron radiators, with the same control as that described for the room in which the first study was made. The unit ventilator in this room was located in the center of the exposed wall, with an auxiliary radiator on each side. The room was vented by a 20 in.

by 20 in. stack, having a 20 in. by 38 in. opening immediately above the floor line in the rear of the room which was left open throughout the study. There was a 10 to 12 mph, SW wind during this study. The occupants of the room had an average body surface area of 17.2 sq ft. This study was conducted in a manner similar to the first study described above, and the results are plotted in Fig. 2.

The third study was made in a room of the same dimensions and equipped with a unit ventilator and two auxiliary radiators having approximately the same location, size and arrangement as the room in which the second study was made. The preparation for and the operation of this study was similar to that of the first two, excepting that at 10:35 a.m., or after the elapse of approximately one-half of the forenoon period, the intake damper was partially opened so as to supply a small volume of outside air. There was a 10 to 12 mph, SW wind during this study. The occupants of the room had an average body surface area of 17.2 sq ft. The results of this study are plotted in Fig. 3.

The pre-arranged conditions under which the three studies were made did not result in as wide a range of air change rates as anticipated. It was hoped that opening the intake damper during the third study would give an air change as high as 15 cfm per person, in order to give an opportunity to observe conditions with air changes ranging from 10 to 15 cfm per person. In order to give data in this range, four extra observations were made in rooms when the normal air change rates were cut down somewhat.

ANALYSIS AND DISCUSSION OF RESULTS OF STUDIES

The CO_2 and water vapor content of the air in an occupied room vary according to a rather complicated relationship between the rate of outside air supply (either positive supply or infiltration) and its CO_2 and water vapor content, the volume of the room under question and the CO_2 and water vapor content of the air contained therein at the beginning of the time considered, and the rates at which CO_2 and water vapor are added to the atmosphere in the room by the occupants. The volume in cubic feet of CO_2 and water vapor contained in the room at any time t minutes after the beginning of the time considered is given by Q in the dilution formula:[2]

$$Q = \frac{V_r}{V_v} V_{gas} \left(1 - e^{-\frac{V_v}{V_r} t} \right) + Q_o e^{-\frac{V_v}{V_r} t}$$

where:

V_r = volume of room in cu ft.
V_v = volume of air supplied in cfm.
V_{gas} = volume, in cu ft, of carbon dioxide or water vapor added to the room per minute.
Q_o = volume, in cu ft, of carbon dioxide or water vapor in the room at the beginning of the time considered or when $t = 0$.
e = base of the Naperian system of logarithms, 2.71828.

This formula applies either while the concentration within the room is being built up or while it is being diluted.

[2] Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 439.

If the volume of the air contained in the room at the beginning of the time considered may be neglected, the relationship between the ventilation rate and the rate of production of CO_2 or water vapor in the room becomes much more simple. The ventilation rate in cfm per occupant is then given by the following formulæ which apply after equilibrium has been reached:

$$R = \frac{C \times \frac{S}{19.5}}{P \times 10^{-4}}$$

where:

R = ventilation rate in cfm per occupant (standard conditions).
C = CO_2 production rate, in cfm per average sized man of 19.5 sq ft (standard conditions).
S = area of average subject, in sq ft.
P = increase in CO_2 concentration, in parts per 10,000.

or,

$$R = \frac{M \times \frac{S}{19.5}}{G \times w}$$

where:

R = ventilation rate in cfm per occupant (standard conditions).
M = moisture production rate in grains per minute per average sized man of 19.5 sq ft (standard conditions).
S = area of average subject, in sq ft.
G = increase in moisture content in grains per lb of dry air as actually determined from wet- and dry-bulb readings and psychrometric chart.
w = weight of air in lb per cu ft (standard conditions).

If equilibrium has not been reached, the application of these formulæ to the conditions in the enclosed room will give a theoretical ventilation rate which would, after the elapse of infinite time, give precisely the same conditions. Hence, the application of these formulæ gives an equivalent rate of air supply. They were used in calculating the "equivalent air supply in cubic feet per minute per occupant" plotted in the upper part of Figs. 1, 2 and 3. The equivalent air change rate based upon CO_2 was computed from: the CO_2 concentration observed in the room and plotted in the lower part of the charts, a CO_2 content in the incoming air of three parts in 10,000, and a rate of CO_2 production per occupant equal to 0.00983 cfm for an average sized man having a body surface area of 19.5 (as found in an earlier laboratory study)[3], and the average surface area of the occupants of the rooms as computed from the school records for the different class periods studied.

The equivalent air supply as plotted in the upper part of the charts, "Computed from the moisture content," was based on a moisture content of the incoming air equal to that found in the outside atmosphere during the period of the study, and the moisture content of the air in the rooms during the studies. In calculating the air change from the moisture content, care must be observed in assuming the proper rate of moisture dissipation from the body, based upon the body surface area of the occupants and the dry-bulb tempera-

[3] Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 245.

ture of the air in accordance with earlier laboratory studies.[4] In this earlier study, it may be of interest to note that an average sized man, having a body surface area of 19.5 sq ft, gave off 11.5 grains of moisture per minute in an atmosphere with a dry-bulb temperature of 70 F, 14.6 grains when the temperature was 75 F, and 18.0 grains per minute when the temperature was 78 F.

The actual equivalent air change per room, based upon the above assumption, is given by the equivalent air supply per occupant as plotted in the upper part of the chart, multiplied by the number of occupants indicated in the middle part of the charts. For most conditions when the occupancy of the room had been

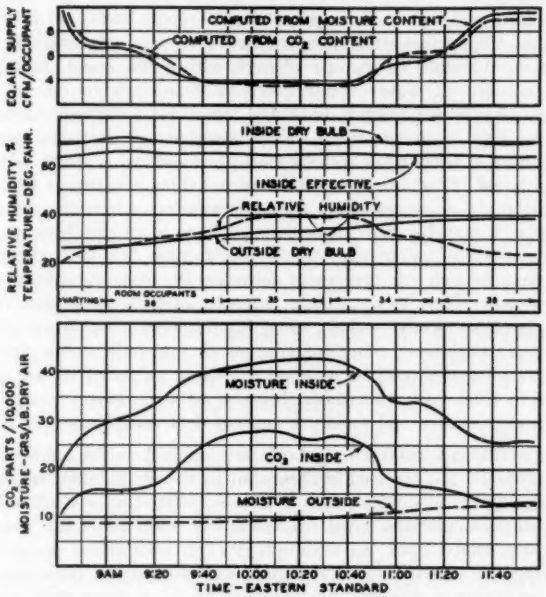
EQ. AIR SUPPLY CFM/OCCUPANT
COMPUTED FROM MOISTURE CONTENT
COMPUTED FROM CO_2 CONTENT
RELATIVE HUMIDITY % TEMPERATURE - DEG. FAHR.
INSIDE DRY BULB
INSIDE EFFECTIVE RELATIVE HUMIDITY
OUTSIDE DRY BULB
ROOM OCCUPANTS
CO_2 - PARTS / 10,000 MOISTURE - GRS/LB. DRY AIR
MOISTURE INSIDE
CO_2 INSIDE
MOISTURE OUTSIDE
9AM 9:20 9:40 10:00 10:20 10:40 11:00 11:20 11:40
TIME - EASTERN STANDARD

FIG. 3. RELATION BETWEEN AIR CHANGE IN A CLASSROOM AND OCCUPANCY. CO_2 PRODUCTION AND INCREASE IN MOISTURE CONTENT. POSITIVE AIR SUPPLY. NO SOLAR RADIATION

constant for a considerable length of time so that equilibrium was approached, air changes ranging from 1 to 1½ per hour were indicated. In the second study all of the occupants with the exception of one observer left the room promptly at noon. The room was then closed and analyses of the CO_2 in the air and other observations were continued for the following hour while the concentration of CO_2 and water vapor decreased due to dilution. The application of the dilution formula given above to this part of the CO_2 curve gives an air change of 1.2 per hour for the room, which is in close agreement with a change of 1.18 based upon an average equivalent air supply during the last hour before noon of 4.24 cfm per person, and a room occupancy of 40 persons.

[4] Loc. Cit. See Note 3.

It will be observed that the equivalent air supply calculated from the CO_2 and moisture content within the room never agreed, although the disagreement is not great in the first or third studies. This disagreement is probably accounted for by dehumidification of the atmosphere within the room, either by condensation on cold surfaces or moisture absorption by other surfaces. In the first study there was a small amount of moisture condensation on the windows, and in the second study there was considerable condensation. During the time when the moisture content in the third study was highest a small amount of condensation appeared on the windows, which disappeared quickly after the rate of air supply was increased by opening the intake damper at 10:35 a.m. Undoubtedly, the reversal in the equivalent air supply indicated between 10:50 and 11:20 a.m. during this study is accounted for by re-evaporation. In applying the dilution formula to the air change within the room, based upon the diminishing content of water vapor after 12:00 o'clock in the second study, an air change of 0.91 per hour is indicated, against 1.2 changes based upon the diminution in CO_2 concentration. This discrepancy may be accounted to re-evaporation of condensation collected on the windows prior to 12:00 o'clock. The probability of reduction or addition in moisture content of the air due to dehumidification or re-evaporation under conditions when the window glass or other surfaces may be at or near the dewpoint would indicate a preference for accepting the equivalent air supply based upon the CO_2 determinations. However, the approach of agreement between the determinations when condensation was least likely to be a factor substantiates the deduction of a former Research Laboratory study [5]—that, when humidification or dehumidification is not a factor, variations in moisture content or wet-bulb temperatures of the atmosphere in a room should be as good or better an index of air change and air distribution than CO_2 variations, because of the greater ease with which the wet-bulb temperature and moisture content determinations may be made.

The dilution formula referred to above was applied to the building up of the moisture content of the air within the room in the first study, based upon the number of occupants and other data. This is plotted as a broken line curve, "moisture inside-calculated," given in the lower section of the chart, Fig. 1. This curve was based upon an assumption of a continuous occupancy of 34 persons during the entire time, and a moisture dissipation from the occupants based upon an average dry-bulb temperature of 78 F from 8:30 a.m. to 9:35 a.m., and a dry-bulb temperature of 75 F thereafter. While the disagreement between this theoretical curve and that giving the results of actual observations is not great, they do indicate a considerably greater rate of building up of moisture content during the first hour than would be based upon the theoretical considerations and assumptions made in the calculated curves. The assumption of a continuous occupancy of 34 persons beginning with the time of 8:40 a.m. was probably too great for the first ten or fifteen minutes of the period. In spite of this fact, however, the curve giving the actual determinations in moisture content rose more rapidly than the theoretical curve. The first observation on which moisture content was based gave a value just equal to the theoretical at 8:56 a.m., with a continued high rate of building up thereafter. This may be accounted for by the fact that the children entering the room had previously

[5] Indices of Air Change and Air Distribution. by F. C. Houghten and J. L. Blackshaw, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 261.

been walking or exercising, thereby having in their clothing an excess amount of perspiration available for evaporation.

The first study was made on a cold day with the expectation that there would be no tendency for the rooms to overheat, or with the expectation that heat would have to be supplied in order to maintain the desired indoor temperatures. This expectation was not realized, however, as indicated by the rapid rise of the dry-bulb temperature after the occupants began to enter to a maximum of 79 F, in spite of the fact that only a very small amount of steam was supplied to the auxiliary radiator in the beginning of the period, which was all cut off as soon as the rise was noted. The temperature was high throughout the entire forenoon, never falling below 74 F. The high dry-bulb temperature taken together with the high moisture content gives an effective temperature range from a high of 71 F to a low of about 68 F. The room was therefore uncomfortably overheated throughout the test. The overheating was later accounted for by the observation that this room was located over the heating plant in the basement, and the added fact that solar radiation was effective through the glass throughout the forenoon. Because of the overheating experienced in this test, a temperature somewhat below normal or approximately 70 F dry-bulb, or 64 to 65 F effective temperature, was maintained during the second study.

This second study, made on the opposite side of the building, was not subject to solar radiation; neither was it subject to unusual heating from below. A small supply of steam was required in the auxiliary radiators throughout the forenoon in order to maintain the desired temperature. This heat requirement diminished to practically nothing during the last hour of the forenoon, indicating that the heat dissipated by the occupants was approximately equal to the heat loss from the room, with an outside temperature of 40.5 F and with no solar radiation entering through the windows.

During each of the three studies, three observers, the teacher of the particular room in which the study was made, and at various times other observers, including the principal of the school, made observations on the ventilation condition within the room, both as regards the general feeling of freshness and presence of odors, upon entering the room and after spending some time in the room.

In judging the condition of odors in the room studied, each observer recorded his judgment as (1) good—meaning that no pronounced odor was noticeable upon entering the room from the corridor of the building; (2) noticeable but not objectionable—meaning that a pronounced odor was noticeable, but that the observer did not judge it objectionable; (3) moderately objectionable—meaning that a pronounced odor was noticeable which was somewhat, though not positively objectionable; and (4) very objectionable.

The sense impression of odors is largely a relative reaction on the part of a person. Hence it should be noted that the above reactions were largely relative to the odor level in the corridor. However, since no noticeable odor was apparent upon entering the corridor from the outside, they may for practical purposes be considered a fair measure of the conditions of the room. It is a well recognized fact that a person may educate his sensitiveness to small variations of odors. No effort was made on the part of the observers to develop a special technique or a fine sense of differentiation. The observations recorded were therefore little different from what would be apparent to one entering such a

room without making an effort to detect the slightest perception of an odor. This procedure, dictated by the short time allowed for the study, would therefore seem to give values of practical value in application.

As stated above, there was no noticeable odor in the corridor upon entering from the outside. Neither was there noticeable odor upon entering any classrooms in the building normally ventilated with approximately 30 cfm of air per person as determined by anemometer readings of velocity.

Each observer gave his judgment of the condition of odor by number as defined above. These numerical judgments by the different observers for a given condition were averaged and are recorded in the last column of Table 1,

TABLE 1. ODOR IMPRESSION FOR DIFFERENT RATES OF AIR CHANGE

STUDY NO.	TIME	DRY BULB TEMP.-°F	EFFECTIVE TEMP.-°F	EQUIVALENT AIR SUPPLY CFM	1-GOOD 2-NOTICEABLE BUT NOT OBJECTIONABLE 3-MODERATELY OBJECTIONABLE 4-VERY OBJECTIONABLE
1	9:20AM	78	71	6.8	3.7
1	10:00	75	68	4.1	4.0
1	10:45	74	68	2.8	4.0
1	11:30	73	68	3.1	4.0
2	9:05AM	68	64	8.0	2.5
2	9:17	70	65	6.8	3.0
2	9:35	68	64	5.3	3.2
2	10:40	70	65	4.0	3.7
2	11:10	70	64	5.6	4.0
2	12:NOON	71	66	3.3	4.0
3	8:44AM	69	64	10.0	2.0
3	8:50	70	65	7.5	3.0
3	9:40	70	65	4.0	3.7
3	10:42	70	66	3.8	4.0
3	11:07	70	65	6.4	3.5
3	11:29	70	65	7.4	3.2
3	11:50	70	65	9.0	2.5
Ex.		70	65	12.2	1.7
Ex.		71	66	14.0	1.2
Ex.		71	65	13.3	1.2
Ex.		71	66	14.2	1.5

which also gives other data on each condition. The number of the study in which the observations were made is given in the first column. The extra, short-time observations made after the three half-day studies were completed are indicated as study "Ex."

The average odor intensity registered by the several observers is plotted against air change in cfm per person in Fig. 4. Different symbols are used for the different studies. A somewhat worse condition is indicated for the same air change in the first study, when the temperature was high; however, there are too few points to draw a definite conclusion in this respect. While there is considerable spreading out of the points in Fig. 4, a rather definite relation

between odor intensity and air change is indicated. Any condition studied with less than 5 cfm per person is shown to be positively objectionable.

If an odor intensity classed as "noticeable but not objectionable" is taken as the lower safe limit of air change, then an air change of from 10 to 11 cfm per person is indicated as necessary. From the observations made in this study, this air change should insure freedom from odors which are noticeable to one not trained in observing them and not particularly looking for their presence. For economy in ventilation, a level midway between (2) and (3), or midway between "noticeable but not objectionable" and "moderately objectionable" may be acceptable. This would permit a reduction in air change to approximately 8 or 9 cfm per person.

It should be emphasized that these deductions apply only to children of junior high school age, or those having an average body surface area of 17.0 sq ft. Since all metabolic processes are assumed to be proportional to body surface area, it is safe to assume that odors may also obey this law, and that when

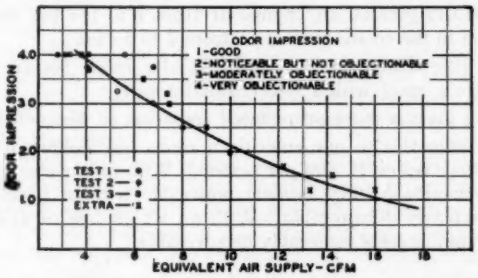

FIG. 4. RELATION BETWEEN ODOR INTENSITY AND AIR SUPPLY

personal cleanliness is not a factor, air change should vary with reference to the body surface area of the occupants.

Provided the temperature of the room was not too high (above 66 F effective temperature), the observers could not notice the odor in the room after being in it for several minutes. Neither was any other discomfort noticeable. In the first study, however, when the effective temperature rose to from 68 to 71 F, great discomfort was noticeable. Odors were apparently not noticed as such, but the air seemed stuffy and the humidity seemed to be noticeable. The teacher in this room, who had no particular knowledge of air conditions in general and who was not informed concerning changes being made in the room, classified the air as "heavy and oppressive." While the conditions were at their worst as far as temperature was concerned, at about 9:30 a.m., the teacher felt it necessary to leave the room for a while to "get a breath of lighter air." If the temperature had not fallen slightly, soon after this time, it is doubtful if she could have been prevailed upon to continue her class. The children were noticeably restless and uncomfortable. The teacher stressed the fact that several children asked permission to go to the toilet room, when under ordinary circumstances there would rarely have been one such request. Whether this resulted from restlessness and a desire to get away from the uncomfortable

condition, or from a physiological derangement, is not known. While the observers had no particular facts or comparisons on which to base their opinions, they were of the opinion that conditions in the room were much less bearable than would be the case with an equally high effective temperature with a greater supply of outside air. It is well to observe that the room was not wanting in air movement, for the unit ventilator was recirculating the usual volume of room air.

Provided the room temperature was not high, there was no apparent sensation of stuffiness or lack of freshness of the air in the rooms to one in a room for sometime. Particularly in the second study, when an effective temperature of from 64 to 65 F was maintained, the condition felt invigorating and the children seemed alert. The teacher pronounced the condition as very good, except for the sensation of odor upon entering the room.

SUMMARY

1. In a classroom of conventional size, air leakage and the initial room air volume was found to give an air change of from 1 to 1½ per hour, or an air supply per person in the rooms studied of from 2 to 5 cfm.

2. Odors are little noticed by the occupants of a room, but are noticeable to one entering from a space with a lower odor intensity.

3. Overheating gives a more pronounced sensation of discomfort than odors.

4. Odors are noticeable to one entering a room and looking for them when the air change drops below 11 cfm per person. With an air change of approximately 9 cfm per person, they become noticeable without looking for them, though not particularly disagreeable. With an air change of approximately 7 cfm per person, they become noticeably disagreeable.

DISCUSSION

C.-E. A. WINSLOW (WRITTEN): I might be inclined to take some exception to the statement at the beginning of the paper to the effect that recent developments have resulted in complete vindication of the idea that a relatively large volume of air should be circulated in the conventional classroom. Certainly the whole trend of recent work has been to justify the policy of the Society in reducing its minimum requirements from 30 to 10 cfm. The same conclusion appears to emerge from the very valuable study made by Mr. Houghten and his associates and it would appear that the studies at Harvard point to the same conclusion, that about 10 cu ft of fresh air per minute is a desirable air supply for the ordinary classroom. I may add that recent studies made at the John B. Pierce Laboratory of Hygiene have furnished new and striking evidence of the injury to health resulting from atmospheric odors, so that we are well within the limits of conservatism in considering the avoidance of appreciable odors on sanitary as well as esthetic grounds, and in requiring a minimum air change of 10 cfm for this purpose.

I would like also to call attention to the great value in studies of this kind of the Fair and Wells osmoscope which have been found very useful in the John B. Pierce Laboratory in the actual determination of the amount of odor present under a given condition.

J. D. CASSELL AND A. J. NESBITT (WRITTEN): We have carefully examined this study, with the following reactions:

(1) We believe that the last sentence in the third paragraph conveys the thought that there is cooling power in recirculated air. We do not agree that this is the

case, for in order for recirculated air to have cooling power it should be mixed with outside air, assuming of course that no artificial means of cooling is employed.

(2) We do not believe that infiltration should be considered in the determination of the quantity of air to be supplied. The rate of air change for a given classroom will vary with many variable factors, such as change in wind direction, change in wind velocity, and variation in building construction. We believe that the air supplied through infiltration should be considered as a factor of safety.

(3) We would like to have the ventilation rate found in the fourth paragraph clarified. Should this be 25 to 10 cfm, or 10 to 25 cfm as written?

(4) The tests were conducted in a school accommodating children who apparently were enjoying better conditions of home life than those in many other schools. It would, therefore, appear that the conditions were not representative of those found in many schools located in other districts. It does not seem logical to determine the rate of ventilation under conditions so favorable to a low air volume in the elimination of odors. Insofar as this test was conducted under conditions favorable from an odor standpoint for a reduced air supply and the students in grades from 7 to 10, it probably does not represent a true cross-section.

(5) What volume of air does the term "rate of air circulation" found in the 10th paragraph refer to?

(6) In regard to the statement that the preparation for and the operation of the third study was similar to that of the first two, it should be noted that in the first study the vent was closed, while in the second study the vent was open. What was the position of the vent in the third study? Would it not be desirable to state the conditions for each study?

(7) We assume that these three studies were made with a constant volume of air circulated by the unit ventilator. Why, therefore, was it not possible to give a rate of ventilation of approximately 30 cfm per person during the third study?

(8) In light of the short amount of time allowed for the study it does not seem advisable to make the statement that the quantity of air may be acceptable for the sake of economy at a volume of 8 or 9 cfm per person. The study clearly indicates that the effect of solar heat and the heat of metabolism is such that on the average day the problem is one of heat removal rather than the addition of heat to the room. We note that for an outside temperature of 38 F and with no solar radiation entering through the windows, the heat dissipated by the occupants was approximately equal to the heat loss from the room.

Table 1 shows that during the second study the equivalent air supply varied from 3.3 to 8 cfm. Increasing this air supply to a minimum of 10 cfm per person would require very little fuel for warming this additional outside air volume. It should be noted that there was no effect from solar heat at the time this second study was being conducted. Rooms exposed to the influence of solar heat would require from 10 to 11 cfm per pupil of outdoor air for cooling with an outside temperature of approximately 38 F. The reference made to a reduction of air volume to 8 or 9 cfm for economy quite logically raises the question of how much economy does such a reduction effect. We are of the opinion that the economy is so small as not to justify reducing the minimum outdoor air volume to the low level which it is suggested may be acceptable.

In the summary under item 4, the statement is made that with an air change of approximately 7 cfm per person odors become noticeable and disagreeable. Based on this fact it would not seem advisable to suggest the reduction in air change to 8 or 9 cfm per person. It does not seem logical to draw the line so finely by saying that 8 or 9 cfm may be acceptable, while 7 cfm per person produces noticeable and disagreeable odors.

(9) The statement appearing in the second paragraph before the summary is confusing and not consistent with other statements made in the paper. It leads one to

believe that so long as the room temperature was properly maintained the observers did not notice the odor in the room. Taken by itself and without an explanation as to the quantity of air being supplied from outdoors while this observation was being made, one might well believe that the report indicates that odors are a matter of temperature. Is this statement to be taken as meaning that 66 deg effective temperature means no odors or discomfort regardless of rate of ventilation?

(10) The selection of the room for the first study was, we believe, a poor one. It represented the conditions peculiar to a room having an unusual heat gain. Studies of this kind should preferably be conducted in rooms where all unnatural sources of heat gain can be controlled.

(11) The paragraph just preceding the summary should be clarified. It contains the same erroneous impression that one would obtain from other similar statements in the paper to the effect that providing the room temperature was properly maintained, there was no apparent sensation of stuffiness or lack of fresh air. Without clarifying these statements how can they be reconciled with other statements in the same report, such as that contained in item 4 of the summary?

Referring to the summary, item 1 should be qualified, limiting this statement to a room or rooms such as those tested having one exposure, and subject to the conditions existing at the time of the test. Item 2 should be qualified as to the intensity of odors, and item 3 qualified as to the intensity of odors compared.

In conclusion, we do not believe that these data should be released until a more thorough study has been made. The fact that this report is based upon studies limited to observations over a period of a little over 3½ days is bound to cause criticism by reason of the importance of the study. The prime object of the study is indicated by its title, Classroom Odors with Reduced Outside Air Supply. The essence of the study is lost in the long explanation of the method used to determine the rate of outside air supplied. The method has merit so far as determining the rate of infiltration, but a more direct method could have been used for determining the volume of outdoor air supplied through the unit ventilator with the same degree of accuracy.

Before using these data we recommend:

(a) That studies be conducted in several classrooms in a school building or buildings located in close proximity to mill districts.

(b) That such studies when conducted should not be confined to children of grades from 7 to 10, but should include both older and younger students.

(c) That tests should be conducted over a longer period in another building in classrooms of approximately the same size and with students of approximately the same age and habits for the purpose of checking the results against those reported in this paper.

(d) Fig. 4 shows that for an equivalent air supply of 15 cfm per pupil, the odor impression was rated as good. Should further studies verify these findings, then it is recommended that condensate readings be taken to determine the actual economy that is to be achieved by reducing the air volume to 10 cfm per pupil, which on Fig. 4 has an odor impression rating noticeable but not objectionable.

J. H. WALKER: I would like to raise objection to the statement in the summary which states that "odors are noticeable to one entering a room and looking for them when the air change drops below 11 cfm per person."

The authors probably intended this statement to apply only to the school under test but the implication is that it is generally applicable. I am sure that we have all entered classrooms in which odors were present even with a much higher rate of air renewal. The odor intensity is greatly affected by the social level of the occupants and the conclusions reached in these tests should be correspondingly restrictive.

J. H. VAN ALSBURG: The papers presented by F. C. Houghten on this subject certainly imply that all previous work that has been done on classrooms, as well as these papers, are what might be called only the beginning of such an investigation because of the many variables that enter into such a series of tests. In connection with this it seems that the angle of discharge from the heater, as well as the velocity, materially affects the room conditions. Therefore, the grille should be of sufficient depth to prevent vision into the cabinet, and the bars should be close enough together so that the deflection is toward the middle of the room, because with the usual high ceilings found in classrooms, the air can be discharged a greater distance without striking an object. Naturally, the longer the air throw the more readily we can expect temperature equalization within a room.

F. C. HOUGHTEN: In the discussion of the paper, reference is frequently made to the inadequacy of the results of three tests for establishing definite conclusions concerning the desired volume of air change in classrooms. The authors are in accord with this point of view. As frequently stated in the paper, the results present evidence collected under the particular conditions specified, which should be of value in formulating ventilation requirements, but is not adequate in itself for the establishment of final conclusions concerning the ventilation requirements of all school buildings.

In interpreting the results, care must be exercised in differentiating between circulated air and supply of outside air. While the supply of from 10 to 15 cu ft. of outside air appears to be adequate, there is argument for the circulation of a larger volume, the balance of which may be recirculated air.

Cooling power of the air, as referred to in the paper, is not measured by its temperature alone, but by all of the qualities of the air which affect the removal of heat from a surface in contact with it, thus including air movements as determined by the Kata thermometer.

The frequent criticism of the discussion of odors in the paper may be clarified in most instances by the difference between a person's sensitivity to odors upon entering a stuffy room and after being confined to the room for some time.

No. 1016

CLASSROOM DRAFTS IN RELATION TO ENTERING AIR STREAM TEMPERATURE

By F. C. HOUGHTEN* (*MEMBER*), H. H. TRIMBLE,** CARL GUTBERLET,*** AND MERLE F. LICHTENFELS† (*NON-MEMBERS*), PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

THE relation between the temperature of the air supplied to classrooms and drafts experienced by occupants has received considerable attention by engineers and others interested in school ventilation. Many suggestions have been made for limiting the minimum temperature of the entering air stream. However, there is no unanimity of opinion fixing the limiting value or specifying if the limiting temperature should vary with the volume, velocity, location and direction of the entering air stream.

The study of the relation between drafts and air stream characteristics was recently given to the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS under the guidance of the Technical Advisory Committee on Minimum Temperature and Method of Introduction of Cooling Air in Classrooms. The Pennsylvania School Heating and Ventilating Commission, in developing a proposed code for regulating the installation of heating and ventilating equipment in schools in the State, recognized the problem. While in their code a limiting value of 55 F was specified for an air stream directed upwards and 60 F for an air stream projected horizontally into the room, the question was recognized as a controversial one and was listed by the Commission with other subjects needing further study.

There is no accepted definition available specifying just what constitutes a draft. In general, a draft is accepted as a condition within an occupied space resulting in local chilling of some parts of the body surfaces of the occupants. While the term is generally associated with, and is derived from air movement, any condition of the atmosphere or surroundings which results in local chilling is generally called a draft. If this interpretation is accepted, then a draft may

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result from a local, low air temperature area within an occupied space, the movement of air against the body so as to give a local chilling effect, or radiation from a part of the body to cold walls or other cold surfaces, or any combination of these three effects. It is doubtful if a person can distinguish between local chilling of a portion of the body surface by any of these three causes.

The study was carried on by the Research Laboratory during the last half of the past heating season, in the Woodlawn Avenue School of Munhall, Pa.

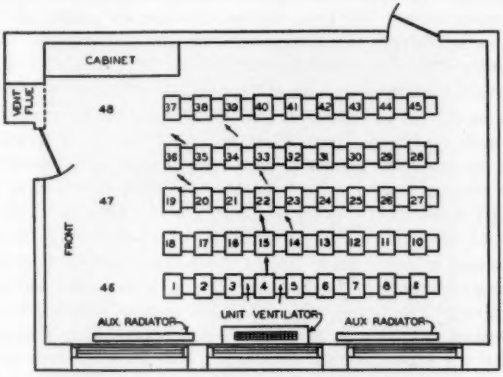

FIG. 1. PLAN VIEW OF ROOM 301 AND PREVAILING AIR STREAMS, SHOWING OBSERVATION STATION NUMBERS USED IN ALL TESTS

FIG. 2. VERTICAL SECTION OF ROOM 301 SHOWING PREVAILING VERTICAL AIR STREAM

A few additional observations were made late in the heating season in the Shaler Township High School, near Pittsburgh. The Woodlawn Avenue School is modern and contains 20 classrooms and the usual arrangement of auditorium, gymnasium, and other space devoted to group activities. Each classroom in this school is heated and ventilated by auxiliary radiators and a unit ventilator. In most rooms the unit ventilator is under a window in the center of the outside wall as shown in Figs. 1 and 2. In these rooms two direct cast-iron radiators are located under windows, one at either side of the unit. In a few of the rooms the unit is located at one side, and a single auxiliary radiator at the other side of the center of the room.

The room temperature is maintained by thermostatic control so arranged as to, first, cut off the steam supply to the auxiliary radiators, and later, to open the by-pass damper, around the heating unit within the unit ventilator, in order to avoid overheating. The units are equipped with dampers controlled from the janitor's quarters in the basement, allowing all units in the building either to take in all outside air or to recirculate room air entirely. Each unit ventilator is wired for low, intermediate and high speed operation, any one of which can be selected by a switch. In the normal operation of the building the units are always operated at low speed during the heating season, giving an average air velocity of approximately 500 fpm, and an average air supply of approximately 1200 cfm, calculated from the outlet velocity and grille area. They are only run at intermediate or high speed during mild weather, when greater air volume is needed to avoid overheating. All rooms are provided with room vent flues.

Classrooms in this school are approximately 22 ft wide from the corridor to the exposed wall, and vary from 28 ft to 46 ft, 10 in. in length. Very few of the corner rooms have a second outside exposure, since most of this space is devoted to stairways and other purposes. In no case did a room have a second exposure containing a window. Additional characteristics of the rooms and operation of the units are given in an earlier Laboratory report.[1]

A few observations were made late in the heating season in a room equipped with a late model unit ventilator built by the same manufacturer. Besides having the features mentioned above for the earlier models, it was equipped with a thermostatic element in the outlet air stream which actuated the steam supply valve to the heater within the unit. This gave greater ease of control of the inlet temperature during a test. The new model gave a higher outlet velocity, but since the grille opening was smaller, the volume of air delivery was not very different.

A few tests were also made in the Shaler Township High School, near Pittsburgh, where rooms equipped with unit ventilators made by a different manufacturer were studied. These units operated about the same as those installed in the Woodlawn school, the only difference being that the discharge grille was set at an angle of about 10 deg with the horizontal, and the unit had no means for speed control except by changes in the wiring.

Most observations were made with normal room occupancy and with the heating and ventilating system operating in accordance with usual practice in this school. In order to avoid having the thermostat change conditions during a survey, means were made available whereby the steam supply to the auxiliary radiators could be controlled by hand operation. Also in some tests, in order to insure continuous operation under the same conditions for a sufficiently long period, the by-pass damper was blocked in the desired position and the steam supply to the auxiliary radiators was controlled by hand. Results obtained under unusual operation of the system were checked by similar tests with normal thermostatic operation whenever possible.

The procedure followed during a survey in any room was to maintain the desired temperature in those portions of the room unaffected by drafts while Kata thermometer and dry-bulb temperature readings were made throughout the room. For this purpose, 12 of the standards shown in Fig. 3 were placed

[1] Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935, p. 279.

on the desks, seats, or the floor of the room. A few strands of silk floss attached to the standards served to give an immediate visual indication of the locations of points of considerable air movement. Actually, the silk floss proved to be more sensitive than necessary, since it indicated air movements as low as 5 or 10 fpm. Greater velocities gave more violent movement of the floss. Sensitive thermometers were also attached to the standards.

Kata thermometer observations were then made at the points of more violent air motion by hanging the instruments on the standards and simultaneously observing the air temperatures and Kata cooling rates. The Kata cooling

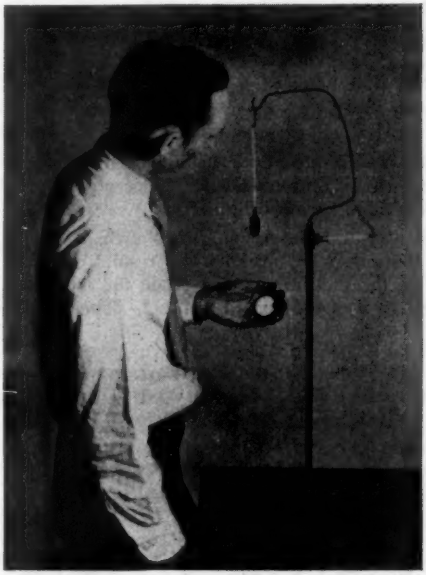

FIG. 3. MAKING OBSERVATIONS OF KATA COOLING RATE AND DRY-BULB TEMPERATURE

rate and the constant for the instrument gives the Kata cooling power directly, or these values and the dry-bulb temperature of the air at the same point gave the air velocity. The dry Kata cooling power in millicalories per square centimeter per second is given for a range of temperatures and air velocities in Table 1.

Simultaneous observations of air temperature throughout the room were made. The observers also made observations of the draft sensation they experienced while seated in the locations in which various air temperatures and velocities, or dry Kata cooling powers, were observed. In some instances the sensations observed at these points by the children and teachers were also recorded.

Since only two Kata thermometers were available for the study, and since

TABLE 1. SHOWING RELATION BETWEEN DRY KATA COOLING POWER AND THE VELOCITY AND TEMPERATURE OF THE AIR

	KATA COOLING POWER FOR KATA AIR VELOCITIES IN FEET PER MINUTE									
TEMP. F	90 Ft	80 Ft	70 Ft	60 Ft	50 Ft	40 Ft	30 Ft	20 Ft	10 Ft	0 Ft
74	6.13	5.89	5.64	5.39	5.10	4.80	4.47	4.13	3.77	3.40
73	6.41	6.16	5.90	5.61	5.31	5.00	4.65	4.30	3.90	3.56
72	6.72	6.45	6.17	5.87	5.55	5.22	4.86	4.50	4.10	3.71
71	6.93	6.66	6.37	6.07	5.75	5.41	5.06	4.70	4.32	3.95
70	7.19	6.92	6.62	6.32	6.00	5.65	5.30	4.93	4.53	4.15
69	7.45	7.17	6.86	6.55	6.21	5.85	5.50	5.11	4.72	4.33
68	7.63	7.43	7.12	6.79	6.44	6.08	5.70	5.30	4.89	4.48
67	7.96	7.66	7.34	7.00	6.65	6.28	5.90	5.50	5.08	4.67
66	8.22	7.92	7.60	7.25	6.89	6.50	6.11	5.70	5.27	4.84

an average of 3 min was required for making a single observation, complete simultaneous surveys of air movements were not possible, but had to be extended over a considerable period of time.

The plan view of a typical classroom, 301, shown in Fig. 1, gives the location at which observations were made. The results of three partial surveys made in this room on three different days, when the incoming air temperature ranged from 37.5 to 40 F, are given in Table 2. The temperature, air move-

TABLE 2. GIVING DRY KATA COOLING OBSERVATIONS AT MOST DRAFTY POINTS OBSERVED IN A TYPICAL ROOM, 301, WITH UNIT DISCHARGE TEMPERATURES RANGING FROM 37.5 F TO 40 F

		KATA OBSERVATIONS											
		45 in. Above Floor			35 in. Above Floor			18 in. Above Floor			6 in. Above Floor		
Date	Station	Temp. F	Velocity Ft/Min	Cooling Power	Temp. F	Velocity Ft/Min	Cooling Power	Temp. F	Velocity Ft/Min	Cooling Power	Temp. F	Velocity Ft/Min	Cooling Power
1/22/35	46	71.0	17	4.58	71.0	22	4.76	72.0	28	4.80	70.0	30	5.30
1/22/35	47	70.5	14	4.57	70.0	18	4.84	69.5	12	4.70	69.5	15	4.82
1/22/35	48	70.5	24	4.94	70.0	35	5.14	69.0	29	5.24	68.0	22	5.38
1/22/35	3	68.0	73	7.22	..	..	..	..	..	..	..	..	..
2/ 5/35	2	71.5	22	4.65	71.0	24	4.84	71.5	30	4.96	70.0	28	5.22
2/ 5/35	20	70.5	24	4.94	71.0	20	4.70	68.0	35	5.88	66.0	38	6.43
2/ 7/35	35	71.0	18	4.63	70.5	25	4.99	69.0	30	5.49	67.0	35	6.08
2/ 7/35	4	67.0	82	7.72	67.0	50	6.64	69.0	30	5.49	68.0	30	5.70
2/ 7/35	15	69.0	40	5.84	68.0	47	6.34	68.0	28	5.63	68.0	20	5.30
2/ 7/35	22	70.8	21	4.78	70.0	23	5.03	69.0	32	5.57	68.0	33	5.82
2/ 5/35	40	71.0	7	4.21	70.5	0	4.06	70.0	10	4.53	69.0	15	4.92
2/ 5/35	24	70.0	30	5.30	70.0	12	4.61	69.5	28	5.32	68.0	22	5.38
2/ 5/35	8	70.8	12	4.44	71.0	21	4.74	72.0	19	4.45	70.5	20	4.80
1/22/35	11	71.0	20	4.70	70.0	19	4.88	69.0	15	4.92	68.0	12	4.98
1/22/35	26	70.0	22	4.98	70.5	17	4.70	69.5	16	4.75	69.0	10	4.72
1/22/35	44	71.5	12	4.28	71.0	8	4.25	70.0	7	4.42	69.5	5	4.45

ment (calculated from the dry Kata thermometer observations), and the Kata cooling power for several of the stations shown in Fig. 1 and for elevations of 6, 18, 35 and 45 in. above the floor are given. The prevailing air currents observed in these surveys are shown by arrows in the plan and vertical views of the room, Figs. 1 and 2. It will be observed that the air delivered by the unit in these surveys was directed upward to or near the ceiling and then turned downward, intersecting the 45-in. level near the first row of seats, and then spread out away from the unit and slightly in the direction of the room vent flue opening. Actually, this prevailing air stream was not easily observed and was only charted after many observations. It is therefore not nearly so apparent or as well defined as indicated in the drawings. It was, however, observed to be more or less present with some variation in all of the surveys made.

Table 2 shows that 45 in. above the floor at station 4, or above the seat immediately in front of the unit, an air velocity of 82 fpm with a dry-bulb temperature 67 F, or a dry Kata cooling power of 7.7, was observed. At the 45-in. level at other points in the room not immediately affected by the entering air, the air velocity ranged from 12 to 30 fpm with a dry-bulb temperature range of from 70 to 71.5 F, or a dry Kata cooling power range of 4.2 to 5.3. The difference between the cooling effect felt by a person when seated in these parts of the room and when seated directly in front of the unit is very great, and anyone subjected to the higher cooling power would classify it as a very undesirable draft.

The normal temperature gradient between the floor and the 45-in. level in the portions of the room unaffected by drafts, or in close proximity to radiators, gave temperatures of approximately 68.6, 69.5, 70.2, and 70.7 F at the 6, 18, 35 and 45-in. levels, respectively. This is in close agreement with the temperature gradients for various rooms in this school reported in an earlier Laboratory paper.[2] In studying the draft conditions in affected portions of the room but at different levels, this normal gradient must be kept in mind. It will be observed that 6 in. above the floor at station 20 an air velocity of 38 fpm with a temperature of 66 F, or a dry Kata cooling power at 6.4, was observed, compared with an average condition in the unaffected portions of the room at this level of 15 fpm with a temperature of 68.6 F, or a 5.0 dry Kata cooling power.

In Table 3 there are tabulated the results of surveys in a number of different rooms with different entering air stream temperatures. The one worst draft condition, only, observed in any survey is recorded, together with the prevailing velocity, air stream temperature, and Kata cooling power.

All rooms in the school did not have the arrangement or relative location of seats, unit ventilator and vent flue opening shown in Fig. 1. The vent flue opening might be located in either the front or rear of the room, and in a few instances, the unit ventilator was not located symmetrically in the center, but either near the front or back of the room. These variations in arrangement are indicated in Table 3.

Although the dry Kata cooling power is made use of throughout the paper, it should be emphasized that it alone cannot be taken as an accurate index

[2] Loc. Cit. See Note 1.

TABLE 3. KATA VELOCITIES AND COOLING POWER FOR DIFFERENT CLASSROOMS AND OPERATING CONDITIONS

DATE	ROOM NUMBER	STATION NEAREST UNIT	LOCATION OF VENT	WORST DRAFT CONDITIONS FOUND IN EACH TEST																								
				UNIT OPERATED AT LOW SPEED								UNIT OPERATED AT INTERMEDIATE SPEED								UNIT OPERATED AT HIGH SPEED							
				TIME	OUTSIDE AIR TEMP F	UNIT DISCHARGE TEMP F	STATION NUMBER	HEIGHT ABOVE FLOOR IN INCHES	KATA OBSERVATIONS			TIME	OUTSIDE AIR TEMP F	UNIT DISCHARGE TEMP F	STATION NUMBER	HEIGHT ABOVE FLOOR IN INCHES	KATA OBSERVATIONS			TIME	OUTSIDE AIR TEMP F	UNIT DISCHARGE TEMP F	STATION NUMBER	HEIGHT ABOVE FLOOR IN INCHES	KATA OBSERVATIONS		
									TEMP. F	VELOCITY FT/MIN	COOLING POWER						TEMP. F	VELOCITY FT/MIN	COOLING POWER						TEMP F	VELOCITY FT/MIN	COOLING POWER
1-22-35	302	9	FRONT	2:45	25.0	35.0	9	45	67.3	67.6	6.94	2:58	25.0	35.0	9	45	69.2	42.5	6.15									
1-22-35	301	4	FRONT	12:29	28.0	38.0	4	45	68.0	73.0	7.22	1:00	28.0	38.0	4	45	68.0	51.0	6.28	11:36	28.0	38.0	5	45	71.3	58.5	5.87
1-24-35	302	9	FRONT	1:13	28.0	36.0	9	45	63.9	103.0	9.60									2:51	18.0	26.0	9	45	68.0	90.0	7.70
1-29-35	305	2	REAR	1:12	36.0	41.0	2	45	67.5	53.0	6.75	1:43	28.5	41.0	14	45	70.3	51.0	5.98	1:52	29.0	41.0	2	45	71.6	61.0	6.00
1-29-35	301	4	FRONT	10:22	30.0	38.5	3	45	67.5	70.0	7.24	11:03	30.0	39.5	5	45	71.7	65.0	6.09	11:54	30.5	39.5	3	45	66.8	44.0	6.46
1-30-35	302	9	FRONT	1:24	21.0	31.0	7	45	62.0	44.0	7.60	11:45	18.0	28.0	6	45	67.1	61.0	7.10	11:23	19.0	28.0	6	45	66.8	74.0	7.30
1-30-35	302	9	FRONT	3:00	26.0	36.0	6	45	67.0	67.0	6.95	2:09	24.0	32.0	8	45	68.5	70.0	6.95	2:44	26.0	33.0	6	45	66.6	81.0	7.80
1-31-35	301	4	FRONT	10:16	29.0	35.0	4	45	65.0	65.0	7.06	10:44	30.0	36.0	4	45	68.0	46.0	6.29	11:07	30.0	37.0	4	45	66.5	75.0	7.15
1-31-35	105	2	REAR	1:22	31.0	40.0	1	45	66.8	75.0	7.55	2:27	31.0	41.0	22	45	71.1	47.5	5.62	2:15	31.0	41.0	18	45	70.7	66.0	6.43
2-1-35	301	4	FRONT	10:14	27.5	37.0	4	35	63.8	55.0	7.05	12:00	34.0	45.5	15	45	72.6	45.0	5.23	12:30	38.5	46.2	41	45	63.8	18.0	5.17
2-1-35	307	4	REAR	2:50	36.0	41.5	3	45	72.3	47.0	5.40	2:26	37.5	50.0	4	45	69.8	31.5	5.40	1:48	36.0	50.0	4	45	68.1	43.0	6.14
2-5-35	301	4	FRONT	10:51	31.0	39.0	20	6	66.0	36.0	6.45	3:00	33.0	41.0	22	6	69.4	39.0	5.32	2:25	33.0	42.8	22	6	69.6	43.0	5.83
2-7-35	301	4	FRONT	11:15	28.0	32.5	4	45	68.0	82.0	7.36									11:55	30.5	40.5	4	45	67.1	62.5	7.05
2-7-35	303	4	REAR	1:05	36.0	41.0	4	45	68.9	35.0	6.71	2:05	36.5	44.0	23	45	66.5	52.0	6.82	1:36	37.0	44.7	13	45	69.3	58.0	6.64
2-12-35	301	4	FRONT	11:39	31.0	41.0	3	35	66.5	43.0	5.96	12:54	38.0	48.0	23	6	67.2	44.0	6.04	1:07	42.0	52.5	21	6	70.0	31.0	5.52
3-12-35	303	4	REAR	11:03	32.5	41.0	4	45	68.5	68.0	6.94																
3-18-35	303	4	REAR									11:13	34.5	55.0	15	45	70.8	38.0	5.40								
3-18-35	303	4	REAR									2:30	50.2	52.5	33	45	70.2	43.0	5.71								
3-19-35	303	4	REAR									10:18	45.0	48.5	3	45	71.7	47.0	5.71								
3-19-35	303	4	REAR									11:50	46.0	50.2	22	45	71.5	43.0	5.40								
3-22-35	303	4	REAR									11:40	51.0	53.0	20	45	71.5	51.0	5.71								
3-28-35	303	4	REAR	11:16	44.0	46.7	15	45	70.2	32.0	5.66	10:19	45.5	46.0	4	45	70.8	43.0	5.96	11:21	44.5	46.5	3	45	67.7	62.0	6.76
3-30-35	AUDITORIUM 208	3	WALL REGISTER EAST									10:52	48.4	53.4	15	45	68.1	33.0	5.79								
4-4-35	SMALL ROOM 209	7	FRONT									9:59	45.0	47.5	11	45	71.2	33.5	5.12								
4-4-35	SMALL ROOM 208	7	FRONT									10:27	46.0	50.5	11	45	71.8	44.0	5.40								
4-4-35	SMALL ROOM 207	7	FRONT									11:16	50.7	52.2	12	45	72.4	38.0	5.01								
AVERAGE ALL TESTS					30.2	39.3			67.2	60.3	6.91		37.6	45.0			70.0	46.0	5.84		31.5	40.4			68.4	38.4	6.32

* SEE FIGURE 1

of a person's feeling of warmth, as shown by earlier Laboratory studies. The effective temperature scale developed by the Research Laboratory has been shown to be an accurate index of a person's feeling of warmth. A 66 deg effective temperature is accepted as the condition for optimum comfort for a person normally clothed and seated at rest indoors during the heating season and with air movements ranging from 15 to 25 fpm. If we accept an air movement of 20 fpm, 66 deg effective temperature will give varying cooling powers ranging from 5.7 at 100 per cent relative humidity, 4.9 at 50 per cent relative humidity, and 3.9 in perfectly dry air. This variation represents the lack of coordination between Kata cooling power and effective temperature as an index of feeling of warmth.

While in general Kata cooling power cannot be accepted as an index of a person's feeling of warmth, variations in the cooling power may, where due consideration is given to relative humidity, be accepted as a very satisfactory index of the relative feeling of warmth and therefore of the sensation of draft. A Kata cooling power of 4.9, resulting from a 66 deg effective temperature, 50 per cent relative humidity, and 70 F dry-bulb, may be accepted for comparison as a most satisfactory air condition. Assuming that a tolerance of approximately 2 deg effective temperature is allowable for reasonably good ventilation conditions, it follows that a variation in Kata cooling power of 0.4 may be accepted, allowing a variation from 4.5 to 5.3.

It should be recognized that a draft has a different effect upon a person than a uniform low effective temperature throughout the room. A person apparently objects less to a lower temperature resulting in an 0.4 increase in Kata cooling power than he does to a localized cooling of the same degree on some parts of the body which is called a draft.

While the conditions available for the study in actual classrooms were not very satisfactory for an accurate study of the relative feeling of discomfort resulting from various air velocities having various air stream temperatures, the following general deductions were arrived at by the observers. An air velocity in the neighborhood of 25 fpm with an air temperature of from 1 to 1½ deg below the prevailing room temperature could be distinguished by its cooling sensation. It would not, however, be classed as objectionable to the observers engaged in the study. Neither was it classed as objectionable by most opinions from children and teachers. In fact, there were more individual opinions expressed that the sensation was pleasant rather than unpleasant. An opinion as to whether such a condition is pleasant or unpleasant is undoubtedly closely associated with the physical condition of the individual. An air velocity in the neighborhood of 40 fpm with an air stream temperature of from 1½ to 2 deg below the prevailing room temperature, or a Kata cooling power of 6.1 compared with 4.9, was decidedly noticeable and was invariably judged by the observers, students and teachers as unpleasant. Air velocities of 50 fpm or greater and with an air stream temperature of 2 deg or more below the prevailing room temperature, or having a Kata cooling power of 1.5 greater than the surrounding air, was always classed as decidedly unpleasant by the observers and children affected by them for more than a few minutes.

As pointed out above, conditions were not satisfactory for arriving at conclusive opinions of the relative effect of drafts on various parts of the body. Since a clear understanding of the effect of various air velocities and tem-

peratures on persons at rest is very necessary for an understanding of the application of air conditioning, it is strongly urged that a careful laboratory study be made of the effects of drafts on persons, particularly with respect to velocity and temperature of the air stream and the portion of the body against which it impinges. There was an apparent tendency for the observers to notice as objectionable low temperature air velocities coming from above and behind more than they did similar conditions affecting other parts of the body including the legs below the knee, while the children occasionally complained of drafts near the floor. This may have resulted from the fact that

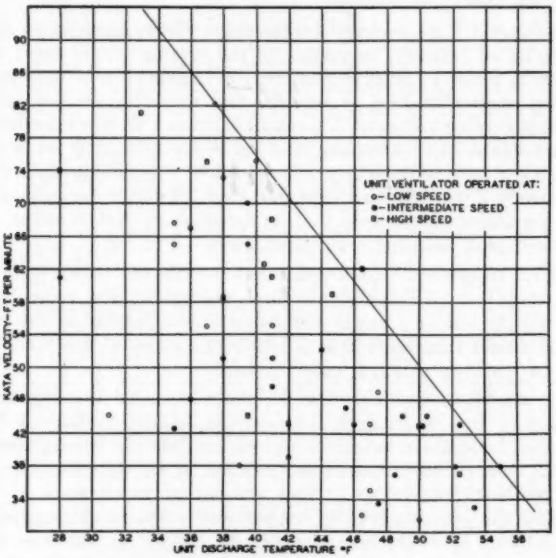

FIG. 4. RELATION BETWEEN UNIT DISCHARGE TEMPERATURE AND VELOCITIES FOR WORST CONDITION IN EACH TEST

the observers were well-clothed adults who had been fairly active before sitting down to make the observations over a short period of time, whereas, the children wore less protective clothing below the knees and had not recently been active. Also, the vitality of the average child may have been less than that of the observers.

Fig. 4 shows how the air velocity observed in the various tests at the most affected points in the room varied with the discharge temperature of the unit. These points were taken from Table 3. In considering them it should be emphasized that the conditions tabulated in Table 3 are the worst observed in the given surveys, which were not always complete and therefore did not always include the worst conditions in the room. This undoubtedly accounts for the wide dispersion of points. It is noticeable, however, that the observed velocity

at the worst points increased with lower discharge temperatures for the conditions and units observed. This demonstrates a fact which may be fairly well accepted—that the excessive air movement observed, resulted from the cold air streams falling quickly to the floor because of their greater density. It follows that the cooling power or draft is doubly affected by low discharge temperatures, since cooling power is increased both by high velocity and low temperature.

Fig. 5 shows how the highest cooling power observed in the several surveys varied with the unit discharge temperature. Again in this figure, the dispersion of points may be accounted for by the fact that the tabulated values in Table 3 did not in all cases represent the worst condition in the room. For this reason, the curves drawn in Figs. 4 and 5 are made to indicate the most

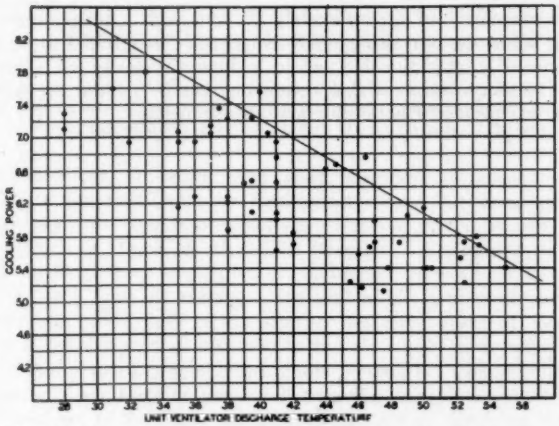

FIG. 5. RELATION BETWEEN UNIT VENTILATOR DISCHARGE TEMPERATURE AND COOLING POWER FOR WORST DRAFT CONDITION IN ALL TESTS

severe conditions which are likely to occur, rather than to fit the group of points.

Figs. 4 and 5 indicate that air velocities as high as 50 fpm and cooling powers as high as 6.1 may be expected to occur in the rooms studied with an entering air stream of 50 F when the average temperature of the unaffected part of the room is between 70 and 71 F. An entering air stream temperature of 55 F may give high localized air velocities of approximately 38 fpm and Kata cooling powers of 5.5. It therefore follows that for the rooms and for the particular heating and ventilating system studied, the temperature of the air stream discharged by the unit must be limited to approximately 55 F, in order to entirely eliminate the probability of localized drafts.

The later model unit used in the Woodlawn school, and the few rooms equipped with another make of unit in the Shaler school were studied when the outside temperature was too mild to obtain very low entering air stream temperatures. These studies do, however, indicate about the same relation

between maximum draft observed and entering air stream temperature as was found for the larger number of rooms studied in the Woodlawn school.

SUMMARY

The following deductions may be drawn from the observations made in the several rooms with the prevailing surrounding conditions.

1. Air movement in the affected portions of the room was found to increase with a lower unit ventilator discharge temperature. The resulting cooling power was also found to increase with a lower discharge temperature.

2. Observations of the worst draft conditions within the rooms during a large number of surveys indicate that undesirable conditions of draft may occur whenever the unit discharge temperature falls below 55 F.

3. An air movement of approximately 25 fpm with a temperature of from 1 to 1½ deg below the prevailing room temperature was found to be noticeable but not necessarily objectionable, while a higher velocity of approximately 40 fpm and a temperature of 2 or more degrees below the prevailing room temperature was decidedly noticeable and objectionable.

No. 1017

AIR SUPPLY TO CLASSROOMS IN RELATION TO VENT FLUE OPENINGS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

THE need for providing vent stack openings in classrooms supplied with air for ventilation by mechanical means has been questioned by many engineers and representatives of school operation and administration. The work of the A. S. H. V. E. Research Laboratory [1] on infiltration demonstrated that a very small pressure difference between the two sides of a wall resulted in a high air change. Studies [2] at the University of Wisconsin have shown that in a few rooms studied the elimination of vent openings resulted in a relatively small decrease in the air supply to a room by a given fan. The Pennsylvania School Heating and Ventilation Commission, appointed to formulate a code governing the installation of heating and ventilating systems in school buildings in the State, debated the question of requiring vent openings in classrooms. Because of the lack of concrete data on the subject, the requirement for vent stacks was included in the code, but the question of their necessity was also included in the report with the suggestion that the engineering profession supply data on which an intelligent disposition of the subject might be based in the future.

In a discussion of the subject the following questions naturally arise:

1. To what extent will the elimination of vent openings in a classroom receiving air from either a central fan system or a unit ventilator reduce the air delivery to the room; or, to what extent will the air pressure in such a room have to be raised above the outside pressure or that pertaining in the corridor in order to maintain the same air supply with the vent opening closed?

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[1] Air Leakage Through Openings in Buildings, by F. C. Houghten and C. C. Schrader, A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924, p. 105. (See also, subsequent reports of studies on infiltration by the A. S. H. V. E. Research Laboratory in the TRANSACTIONS of the Society.)

[2] Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 463.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., June, 1935, by F. C. Houghten.

2. What effect will the elimination of vent openings have on air distribution throughout the room?

3. What effect will the elimination of vent stack openings have on the temperature gradient between the floor and ceiling?

The A. S. H. V. E. Research Laboratory is engaged in a study of the relation between characteristics of air supply to a classroom and drafts in schools in the Pittsburgh district. In connection with this study it was found convenient to make a few observations of the effect of eliminating vent stack openings on the ventilation of classrooms. The study included observations in all classrooms in the Woodlawn Avenue School at Munhall, Pa., mentioned in an earlier Laboratory report,[3] and in 16 of the classrooms in the Shaler High School, near Pittsburgh.

All classrooms in both schools were provided with vent flue openings extending to the outside at the roof of the building. The openings in the Woodlawn school were 20 in. wide by 38 in. high, with the bottom of the opening about 10 in. above the floor line. In most instances the vent flue openings were located in cloak rooms provided with doors extending down to 6 in. from the floor. The cloak room doors were always kept closed during a test. There were no vent openings between the rooms and corridors. The doors were reasonably tight around the sides but had clearances of from ⅛-in. to ½-in. at the bottom. All windows were weatherstripped.

The corridors in the schools were not provided with vent openings, but in order to insure against building up of a pressure in the corridor due to air leakage into them during a test, a corridor window on the floor on which a test was being made and on the side of the building least affected by the wind was opened. The corridors were therefore at atmospheric pressure and the results of the study should be typical of schools in which the corridors are amply vented to allow the escape of air leaking into them from classrooms not having vent openings into flues or into the corridors.

Each classroom was provided with one or two auxiliary radiators depending on the location of the unit ventilator. Where the unit was in the center of the exposed wall there were two auxiliary radiators, one on each side of the unit. Where the unit was at either end of the exposed wall there was a single auxiliary radiator at the other end.

In preparing for a study in any room, an inclined manometer was placed in the room near the door, with one leg connected to a ⅛-in. brass tube extending under the door to the corridor so as to indicate the pressure difference between the inside of the room and the corridor. A calibrated anemometer was placed on the unit ventilator outlet grille at a point near the center which was found to give an approximately average outlet velocity. This anemometer was not moved during the test. In making the study in a few of the rooms, thermometers were placed,—6 in. from the floor, 30 in. from the floor, 63 in. from the floor, 101 in. from the floor, and 6 in. from the ceiling,—in a location in the plan of the room which was half-way between the exposed wall and the corridor wall, and approximately one-third of the way from the front or rear wall to a line through the center of the unit ventilator. In rooms where the unit ventilator was not placed in the center of the exposed wall, this

[3] Classroom Odors with Reduced Outside Air Supply, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and Merle F. Lichtenfels, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935, p. 253.

location was chosen in the end of the room having the greater distance from the front or rear wall to the center of the unit ventilator. Other thermometers were placed on the desks in different portions of the room in order to indicate any variation in temperature at this level.

Observations were taken at frequent intervals of the thermometers giving the temperature gradient from the floor to the ceiling, and the temperature variation at the level of the tops of the desks, the pressure difference between the room and the corridor, and the reading of the anemometer located on the unit ventilator outlet grille, over a period of 20 min or more, with the vent open and the unit ventilator operating as installed. All windows and doors were closed as in normal practice. At the end of this period the vent stack opening was completely closed and the observations were continued during another 20 min interval or longer.

The unit ventilators in the Woodlawn school could be operated at low, intermediate, and high speed by a switch conveniently located in the unit. Most of the units in the school were normally operated at low speed. Observations were made on the effect of closing the vent with both the low and high speeds in all the rooms, and with the intermediate speed also in a few cases. In the Shaler school observations were only made with the unit ventilator running at the speed normally used.

The results of the study in the rooms of the two schools are given in Table 1. The study in the Woodlawn school extended over a considerable period of time and included observations on days with different outside temperatures. The observations in the Shaler school were made on a single day later, with a higher outside temperature.

When the units were operating at low speed in the Woodlawn school they delivered from 914 to 1343 cfm to the different rooms, based upon the measured outlet velocities and grille areas. For these conditions the different rooms showed percentage reductions in the volume of air delivery with the vent closed ranging from 1.7 to 25.9 per cent of the delivery with the vent open; the highest decrease, 25.9 per cent, was found for the library.

With the unit ventilators operating at high speed in the Woodlawn school the air delivery rate was naturally higher, and the percentage decrease with vents closed was greater, ranging from 4.7 to 22.2 per cent of the delivery with the vents open.

The table also gives averages for different conditions pertaining during the different tests in the Woodlawn school, including outside temperatures of approximately 27 F, 35 F and 46 F, the floor on which the rooms were located, the direction of exposure of the rooms, and for the entire series. These data show no consistent variation in reduction of air delivery with any of these variables.

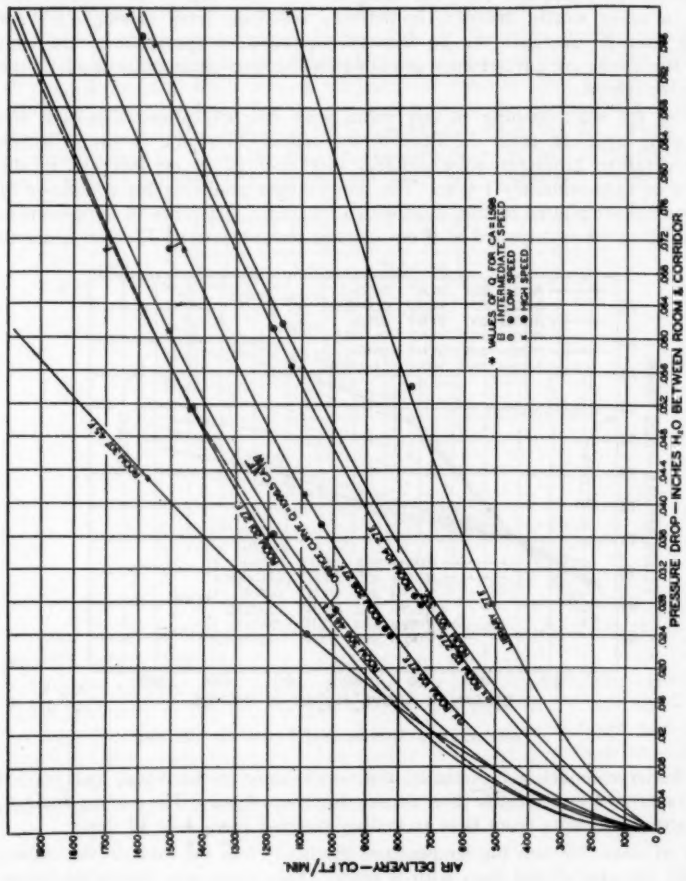
The results of the tests in a few of the rooms are shown in Fig. 1, where the air delivery by the unit ventilator is plotted against pressure difference between the room and the corridor for the two unit ventilator speeds and with the vents closed in the Woodlawn school. The curves show a wide variation in the relation between pressure drop and delivery for the different rooms, but they all show similar characteristics to curves for air flow through orifices. A single orifice curve is plotted as a broken line for a value of $CA = 1.598$.

TABLE 1. RESULTS OF STUDY IN WOODLAWN AND SHALER SCHOOLS

ROOM NO	LENGTH OF ROOM FEET (EXTERIOR)	WIDTH OF ROOM FEET	VOLUME OF ROOM CU FT	NO OF WINDOWS	EXPOSURE	WIND VEL. AND DIRECTION	AIR DELIVERED BY UNIT VENTILATOR						CLOSURE IN AIR DELIVERY	DIFFERENCE BETWEEN TO CORRIDOR PRESS WATER COLUMN	
							VENT OPEN			VENT CLOSED					
							VELOCITY FT/MIN	VOLUME C.F.M.	AIR CHANGES/HR	VELOCITY FT/MIN	VOLUME C.F.M.	AIR CHANGES/HR	WITH VENT CLOSED	VENT OPEN	VENT CLOSED
WOODLAWN AVE. SCHOOL															
101	32'0"	22'4"	8576	3 *	EAST	6SW	L 331	1826	5.76	497	1125	7.87	10.1	.0001	.0435
							H 575	3084	14.47	713	1870	11.76	18.1	.0086	.0685
102	38'5"	22'4"	10296	3	EAST	5SW	L 340	1981	7.70	388	1184	6.91	10.2	.0092	.0610
							H 771	1842	10.56	691	1830	9.90	10.0	.0719	.0992
103	32'0"	22'4"	8576	3	EAST	6SW	L 287	1643	6.70	480	1132	7.92	8.3	.0044	.0633
							H 782	1846	13.03	716	1681	11.77	8.7	.0569	.1503
104	32'0"	22'4"	8576	3	EAST	6SW	L 349	1295	5.06	476	1123	7.86	13.3	.0337	.0584
							H 730	1763	12.33	657	1530	10.84	12.1	.0374	.0933
105	28'0"	22'9"	7352	2	WEST	5W	L			442	1088	6.84			.0610
							H 778	1830	14.07	622	1465	11.67	20+	-.0017	.0706
106	28'0"	22'0"	7392	3	WEST	6SW	L 384	914	7.42	331	833	6.76	8.5	-.0218	.0913
							H 563	1342	10.73	536	1260	10.23	4.7	-.0052	.0350
AVERAGE FIRST FLOOR							L 291	1060	6.33	458	1081	7.69	10.4	.0073	.0480
							H 784	1774	12.65	636	1544	10.99	13.0	.0877	.0848
201	32'0"	22'4"	8576	3	EAST	4W	L 473	1803	8.11	703	1664	8.72	9.1	-.0010	.0551
							H 843	1827	13.44	743	1705	11.52	14.8	.0235	.0705
202	38'5"	22'4"	10296	3	EAST	4W	L 543	1189	7.47	480	1062	6.78	17.1	.0070	.0376
							H 760	1780	10.46	680	1726	8.80	16.3	.0031	.0515
203	32'0"	22'4"	8576	3	EAST	3NW	L 476	1122	7.83	410	969	6.78	15.6	-.0081	.0223
							H 700	1646	11.52	602	1416	9.88	14.1	-.0026	.0680
204	32'0"	22'4"	8576	3	EAST	6NW	L 467	1107	8.02	421	1002	7.69	12.3	-.0119	.0224
							H 709	1842	12.89	648	1725	10.67	17.2	-.0067	.0634
205	28'0"	22'0"	7392	2	WEST	6NW	L			440	1038	8.43			.0512
							H 635	1692	12.11	604	1620	11.53	6.8	.0398	.0620
206	28'0"	22'0"	7392	3	WEST	6SW	L 482	1138	8.40	440	1038	8.43	10.4	.0037	.0376
							H 708	1660	13.47	641	1508	12.24	9.2	.0267	.0707
LIBRARY	44'0"	22'0"	12144	3	WEST	7SW	L 433	1633	2.05	318	737	2.74	28.5	-.0231	.0540
							H 626	1472	7.27	484	1162	5.44	24.6	-.0118	.0827
AVERAGE SECOND FLOOR							L 488	1172	7.52	424	1020	6.97	14.2	.0016	.0363
							H 722	1696	11.64	622	1465	10.11	13.7	.0071	.0674
301	32'0"	22'4"	8576	3	EAST	3W	L 490	1178	8.22	424	1046	7.33	10.8		
							H 760	1898	12.78	632	1520	10.63	15.6		
302	38'5"	22'4"	10296	3	EAST	6NW	L 498	1172	6.83	437	1027	5.98	12.4	.0104	.0722
							H								
303	38'5"	22'4"	8576	3	EAST	7NW	L 534	1392	9.11	482	1136	8.16	11.1	.0131	.0617
304	32'0"	22'4"	8576	3	EAST	7NW	L 436	1069	7.20	428	1011	7.07	1.7	.0183	.0378
305	28'0"	22'0"	7392	3	WEST	6W	L 572	1363	10.90	434	1023	8.32	23.7	.0804	.0838
							H 884	2100	16.06	734	1728	14.03	17.9	.0000	.0828
306	28'0"	22'0"	7392	3	WEST	5W	L 449	1105	8.97	418	990	8.64	10.4	.0065	.0270
							H 706	1660	13.47	642	1510	12.26	9.1	.0256	.0408
307	46'10"	22'0"	12364	3	WEST	5W	L 519	1221	5.93	457	1078	5.25	11.7	-.0003	.0240
							H 803	1890	9.17	646	1520	7.42	15.2	-.0002	.0483
AVERAGE THIRD FLOOR							L 307	1182	8.18	441	1048	7.12	12.2	.0022	.0442
							H 782	1863	13.07	660	1568	11.13	17.1	.0127	.0522
AVERAGE ALL FLOORS							L 368	1181	8.11	442	1043	7.27	12.4	.0063	.0432
							H 726	1765	12.17	649	1528	10.74	15.3	.0185	.0681
SHALER HIGH SCHOOL															
101	*20'2"	22'0"	10348	6 **	SOUTH EAST	7W	498	961	6.58	475	917	5.33	4.6	.0017	.0220
102	32'7"	22'0"	8582	5	SOUTH EAST	6W	693	1318	9.21	623	1202	8.39	8.8	.0017	.0508
103	26'1"	22'0"	6880	4	SOUTH EAST	4NW	588	723	6.30	580	676	5.89	6.9	.0000	.0225
105	35'0"	22'0"	9240	5	SOUTH WEST	5NW	550	1062	6.89	521	1006	6.53	5.3	.0017	.0213
106	26'1"	22'0"	6880	4	SOUTH WEST	6W	680	836	7.28	620	763	6.64	8.8	.0038	.0478
AVERAGE FIRST FLOOR							600	980	7.06	558	913	6.56	6.8	.0017	.0328
201	26'3"	22'0"	6930	4	SOUTH EAST	8NW	716	883	7.65	670	824	7.13	6.7	.0000	.0387
204	26'0"	22'0"	6864	4	SOUTH WEST	7NW	697	857	7.48	630	775	6.77	9.6	.0052	.0440
208	26'4"	22'4"	6951	4	SOUTH WEST	7NW	679	835	7.21	620	763	6.59	9.6	.0066	.0613
210	26'0"	22'0"	6864	4	SOUTH WEST	6NW	550	676	5.91	519	638	5.58	5.6	.0035	.0364
211	33'0"	22'0"	8712	5	NORTH WEST	8NW	470	807	6.24	440	849	6.85	6.4	.0000	.0406
213	26'3"	22'0"	6930	4	NORTH WEST	6NW	600	738	6.39	524	645	5.58	12.6	.0052	.0691
AVERAGE SECOND FLOOR							618	846	6.86	567	748	6.25	8.2	.0034	.0520
302	26'0"	22'0"	6864	4	SOUTH EAST	9NW	787	980	8.56	680	849	7.43	13.4	.0115	.0498
305	26'0"	22'0"	6864	4	SOUTH WEST	8NW	545	670	5.85	495	609	5.32	9.2	.0082	.0578
306	26'0"	22'0"	6864	4	SOUTH WEST	8NW	727	884	7.81	653	803	7.02	10.2	.0069	.0526
308	28'0"	22'0"	7392	4	SOUTH WEST	9NW	779	958	7.78	700	861	6.99	10.4	.0026	.0530
310	38'6"	22'0"	10428	6	NORTH WEST	9NW	547	1247	7.17	505	1151	6.62	7.7	.0005	.0346
AVERAGE THIRD FLOOR							679	950	7.44	609	855	6.69	10.2	.0089	.0497
AVERAGE ALL FLOORS							632	915	7.12	578	839	6.50	8.4	.00346	.0419
AVE. ALL TESTS IN WOODLAWN SCHOOL WITH 27°F OUTSIDE TEMPERATURE.							LOW SPEED	1189			1053		12.8		
							HIGH SPEED	1736			1514		13.5		
AVE. ALL TESTS IN WOODLAWN SCHOOL WITH 35°F OUTSIDE TEMPERATURE.							L	1169			1061		8.2		
							H	1690			1520		18.5		
AVE. ALL TESTS IN WOODLAWN SCHOOL WITH 46°F OUTSIDE TEMPERATURE.							L	1168			1016		14.8		
							H	1778			1525		13.3		
AVE. ALL TESTS IN WOODLAWN SCHOOL WITH ROOMS FACING EAST							L	1127			881		16.3		
							H	1803			1430		13.7		
AVE. ALL TESTS IN WOODLAWN SCHOOL WITH ROOMS FACING WEST							L	1218			1085		10.9		
							H	1864			1681		13.9		

* GLASS AREA PER WINDOW = 44.8 SQ. FT.
** GLASS AREA PER WINDOW = 28.8 SQ. FT.
*** L-UNIT OPERATING AT LOW SPEED
M- " " " INTERMEDIATE SPEED
H- " " " HIGH SPEED

If for this curve an orifice constant, $C = 0.6$, is assumed, a resulting orifice area of 383 sq in. would be indicated, bearing out the findings of the infiltration studies, that the equivalent orifice area of all the avenues of leakage from a


AIR DELIVERY—CU FT/MIN.
PRESSURE DROP—INCHES H₂O BETWEEN ROOM & CORRIDOR
* VALUES OF Q FOR CASE 1 INTERMEDIATE SPEED
□ ● LOW SPEED
x ● HIGH SPEED
ROOM 201 VENT CLOSED
ROOM 201 VENT OPEN

for open and closed vents with units operating at different speeds, as tabulated in Table 1, that the reduction in air delivery due to closing the vent with the unit operating at low speed is considerably less than the increase in delivery with closed vent and the unit operating at the next higher or intermediate speed; in other words, decrease in delivery resulting from closing the vent opening may be overcome by an increase in speed of operation of the unit within the range of adjustments provided in the particular units used in the Woodlawn school.

Closing the vent opening in any room with few exceptions increased the temperature gradient from the floor to the ceiling. For the rooms in which the temperature gradients were studied, this resulted in an increase in the gradient of approximately 1 deg. The temperature gradients for a number of cases studied are plotted in Fig. 2, in which the slight difference in temperatures at the 30-in. level is adjusted to their average temperature of 71 F in order to

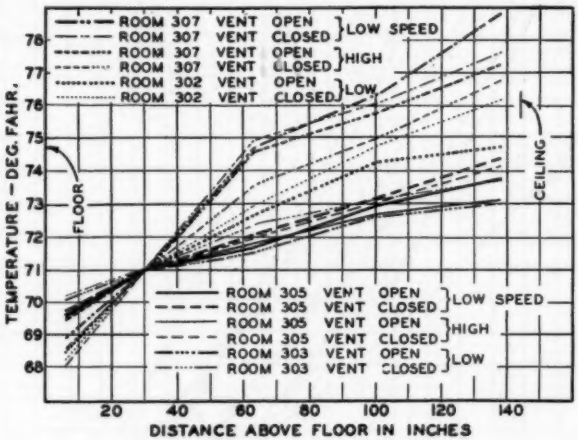


FIG. 2. TEMPERATURE GRADIENTS, FLOOR TO CEILING

give a better comparison. Actually, the temperature at the 30-in. level varied by as much as three-fourths of a degree for these tests. The curves indicate temperature gradients from floor to ceiling ranging from 3 to 10 deg.

Only in room 307 was the temperature gradient from the floor to the ceiling less with the vent closed than with it open. This room was somewhat larger than the average room in the building and had a lower room air change under normal operation of the unit, which undoubtedly accounts for the greater temperature gradient between the floor and ceiling with the vent open for the same unit speed than found for the other rooms. In all cases observed, the temperature gradient from the floor to the ceiling was lower with high speed than with low speed operation of the unit.

While the closing of the vents usually resulted in a 1 deg higher temperature at the ceiling, the resulting difference in the gradient between the floor line and the 30-in. level was usually smaller, and probably not significant. In other

words, the closing of the vent opening did not unduly affect the temperature gradient in the occupied portion of the room; neither did it give a very significant rise in the ceiling temperature. The thermometers distributed throughout the room at the level of the tops of the desks showed no significant variation in temperature correlating with the opening or closing of the vents.

Observation was also made of the effect of the high pressure within the room on the operation of the doors leading to the corridor. Since the doors all opened outward, there was no tendency for them to slam in closing. There was a noticeable tendency for the pressure to force the door quickly open when the latch was released; however, not enough to snatch the door away from a person opening it. With the highest pressure observed in the rooms, the door when unfastened and released did not open with sufficient speed to hit the

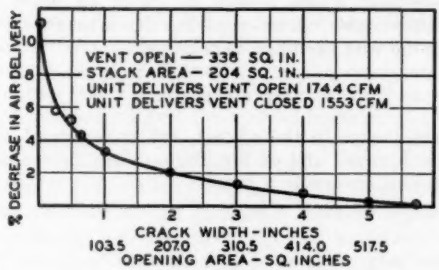


FIG. 3. RELATION BETWEEN DOOR OPENING AND THE PERCENTAGE REDUCTION IN AIR DELIVERY TO THE ROOM WITH VENT CLOSED, FROM THE AIR DELIVERY FOR THE SAME ROOM WITH THE VENT OPEN AND THE DOOR CLOSED

corridor wall. Usually, a door so released opened to about the three-quarter position before stopping.

In order to offer some evidence of the effect which a corridor vent opening either under or in the door would have, a few tests were made in which the pressure drop and air delivery were observed with the vent closed, and with the door opened various distances. The results of one of these tests are plotted in Fig. 3, showing that when the door was opened enough to allow a 2-in. crack the reduction from normal open vent air delivery was only 2 per cent.

SUMMARY

1. The study, made in the 20 rooms of the Woodlawn school and in the 16 rooms of the Shaler School, indicates that, with normal unit ventilator operation and with all windows and doors closed, the closing of the vent opening decreased the volume of air delivery to the room by the unit ventilator from 1.7 to 25.9 per cent of the delivery with the vent open. The average reduction in air delivery was found to be 10.5 per cent for all rooms. This reduction in air delivery shows no consistent relationship with outside temperature, window area, or other room conditions, but seemed to depend more upon individual room characteristics.

2. The reduction in air delivery resulting from closed vents with low speed operation of the unit ventilators could be more than offset by an increase of the unit ventilator speed from low to intermediate.

3. Closing the vent openings usually resulted in a greater temperature gradient from the floor to the ceiling, and a higher temperature of approximately 1 deg at the ceiling. While the closing of the vent openings resulted in a slightly lower floor temperature for the same temperature at the 30-in. level, the drop in temperature was not great.

4. Closing the vent openings did not result in a greater temperature variation throughout the room at the 30-in. level in either an occupied or an unoccupied room, indicating equally good air distribution throughout the room with or without vent openings.

5. For the particular rooms studied in which all doors opened outward, no slamming or undesirable operation of the doors resulted from the pressure in the room when the vent openings were closed.

ACKNOWLEDGMENT

Acknowledgment is due to the officers and management of the Woodlawn Avenue School in Munhall and of the Shaler High School for their valuable cooperation in making the study a success.

No. 1018

HEAT ABSORBING GLASS WINDOWS

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IT is a well recognized fact in the air conditioning field that the sunlight energy coming directly through the windows of a home or office may constitute a very considerable percentage of the total cooling load on the refrigeration equipment controlling the room temperature and general comfort of the occupants, particularly where the thermal insulation of the sidewalls has reduced the heat conduction load to a minimum. On this account, any effective means of reducing this solar energy load without materially interfering with the original purpose of the windows may be of real importance in broadening the application of air conditioning through a material reduction in the peak load and operating cost of the equipment. Special heat-absorbing clear glasses having high light transmission with very little color have been suggested for this purpose and discussed in a number of recent papers.[1, 2, 3] In connection with the use of these glasses as window glasses, questions have arisen regarding the potential economies in the operation of air conditioning equipment; the effect of such windows on the general comfort of the occupants of the room or dwelling; their effect on the fuel bill in winter in case of a permanent installation; and the behavior of the glass under general service conditions. In order to provide some information on these points, a series of tests was carried out during the greater part of 1934, the results of which are summarized in this paper.

Before describing the experiments in detail it may be well to give a brief outline of the process by which heat-absorbing glass is effective in preventing a considerable portion of the solar energy from passing through the window. Sunlight falling on any glass window is partly reflected, partly absorbed, and the remainder transmitted directly into the room. The amount of reflected energy varies with the angle of incidence, being about 8 per cent for clear glass at normal incidence, and insofar as the interior of the building is concerned this reflected energy is effectively lost. The energy that is absorbed by the window heats the glass, which in turn dissipates this heat by conduction, convection, and radiation in all directions, the temperature of the glass meanwhile

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[1] Hobbie, *Transactions, Illuminating Engineering Society*, Vol. 28, 1933, pp. 658-664.

[2] Estey and Miller, *Transactions, Illuminating Engineering Society*, Vol. 30, No. 5, 1935, pp. 403-410.

[3] Shaver, *Refrigerating Engineering*, Vol. 26, No. 3, 1933, pp. 133-136.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June 1935.

rising until the heat lost by this process is equal to that absorbed. Thus, with single windows, in quiet air, approximately one-half of the energy absorbed by the glass finally enters the room through this secondary reradiation process. Now ordinary window glass absorbs only about 4 per cent of the total energy in sunlight, transmitting 92 per cent of the visible light, which constitutes 44 per cent of the sun's total energy. (See Table 1.) Hence the amount of energy entering the room by the reradiation process is very small in comparison with the direct sunlight transmitted through the glass, and likewise there is very little heating of the glass.

In the case of special heat-absorbing glass the selective absorption effect is very much enhanced, with a total absorption of about 53 per cent of the total energy in sunlight and a visible light transmission of 76 per cent. With this marked increase in absorption, the reradiation effect becomes an important factor in transmitting energy into the room; so that while a very desirable heat screening effect is obtained with a single window of heat-absorbing glass, due to the fact that the glass dissipates heat outwards as well as inwards, much

TABLE 1. CHARACTERISTICS OF GLASS USED IN TESTS

GLASS	THICKNESS	VISIBLE LIGHT TRANSMISSION	TOTAL SOLAR ENERGY TRANSMISSION	TOTAL SOLAR ENERGY REFLECTED
Borosilicate Heat-resisting Heat-absorbing plate	3 mm	76%	42%	5%
Ordinary Window Double Strength	3 mm	92%	88%	8%

better results should be obtained through the use of double windows, with ordinary glass in the inner sash, special heat-absorbing glass in the outside sash, and means provided for removing the heat absorbed by the outer sash glass as rapidly as possible. Another consequence of the high percentage of energy absorption in the heat-absorbing glass is that the glass becomes very hot when exposed to intense sunlight so that it may be subjected to very severe thermal stresses, as for example when shadows cover part of the window. Hence the thermal resistance of heat-absorbing glass must also be considered.

TWO TEST HOUSES USED

On account of the continual variation in the energy in sunlight at the earth's surface, duplicate test equipment must be available in order to make a direct comparison of the two types of glazing. Accordingly, two test houses, shown in Fig. 1, of identical design and construction were built on the roof of a 4-story factory building at Corning, N. Y., in a location free from objectionable shadows and where atmospheric conditions were satisfactory. Each house was of wood frame construction with a single insulated and tightly fitting door and two windows which faced the southeast and southwest. The interior dimensions of the houses were 8x8x8 ft and the window sash were each glazed with 16 lights of 12 $\times$ 18 in. glass. Two layers of ½ in. fiber insulating board were used to insulate the sidewalls and three layers of the same insulation protected

the ceiling. The walls and ceiling of fiber insulating board were sized with a heavy coating of shellac to prevent the absorption of moisture.

Each house was cooled by circulating ice water continuously from a central outside storage tank through the cooling coil of a room cooler whose fan was operated by means of a thermostatic relay. The rise in temperature in the water flowing through the cooling coil was measured by a series of 10 differential copper-constantan thermocouples, one set of junctions being placed in the water stream at the point of entry and the other set in the stream at the point of exit. This temperature difference was recorded on a potentiometer recorder. The flow of water through each cooler was measured with a specially calibrated water meter. Thus from the record of the rise in temperature of the water stream and the metered flow of the water, the amount of heat withdrawn in any given time could be determined. In order to keep the temperature of the room as uniform as possible and to insure good thermostatic control, a 10-in.

FIG. 1. TEST HOUSES USED IN HEAT-ABSORBING GLASS EXPERIMENTS

electric fan was mounted in one corner near the ceiling and operated continuously with its blast directed downward across the bimetallic element of the thermostat. The entire electric power used in operating the fans and recorders was measured with an electric meter in each house.

The characteristics of the borosilicate heat-resisting, heat-absorbing glass and the ordinary window glass used in the tests have been given in Table 1. The heat-absorbing glass was 3 mm polished plate of a slightly bluish-green color, but so light in tint that it could scarcely be noticed by an observer standing inside the test house. The ordinary window glass was of double strength drawn sheet.

TESTS CHECKED

Tests were made on every suitable day from July 16 to October 10, 1934. During this period the weather was not very satisfactory for the experiments, and in the entire period only two days could be classified as ideal. The first set of tests was made with double windows, one house having ordinary glass in both inner and outer sash and the other having ordinary glass in the inner sash and heat-absorbing glass in the outer sash. All the outside sash were equipped with slots at the top and bottom which permitted reasonably free circulation of air between the windows. Check tests were made by interchanging

the window combinations on the two houses. The results of these tests are given in Table 2.

SASH VENTILATION

In order to determine the effect of the slots on the outer sash and the resulting ventilation of the space between the windows, one run was made in which the slots in the outer sash of No. 1 house with the combination of ordinary and heat-absorbing glass were alternately opened and closed at the end of every hour, while the slots in the outer sash of No. 2 house with the ordinary glass windows were left open during the entire period. The results are given in Table 3 and show rather definitely the effect of ventilation between the sash.

TABLE 2. RESULTS OF CHECK TESTS WITH DOUBLE WINDOWS

DATE—WEATHER	MAX. OUTSIDE TEMP. F	TIME OF OPERATION HOURS	HEAT ENERGY REMOVED		SAVING DUE TO HEAT-ABSORBING GLASS %
			No. 1 House Btu	No. 2 House Btu	
No. 1 House—Double Windows—Ordinary Glass + Ordinary Glass *No. 2 House—Double Windows—Ordinary Glass + Heat-absorbing Heat-resisting Glass*					
7/16 Sunshine with cumulus clouds	77	8	24810	22540	9.2
7/17 Sunshine with cumulus clouds	86	6	21210	17830	15.9
7/19 Sunshine with cumulus clouds. Quite cloudy after 3 P.M.	99	5	22140	20010	9.6
7/20 Cloudy until 11 A.M. then clear with cumulus clouds	97	3.7	18870	16550	12.3
No. 1 House—Double Windows—Ordinary Glass + Heat-absorbing Heat-resisting Glass *No. 2 House—Double Windows—Ordinary Glass + Ordinary Glass*					
7/22 Clear—some haze—cloudy after 4 P.M.	83	3.7	13830	16990	18.6
7/23 Clear with few cirrus clouds	86	6	17200	20730	17.0
7/25 Hazy—sky filling with clouds—weather unsatisfactory until noon	92	7	28990	32360	10.4
7/30 Hazy with few cirrus clouds in morning—cumulus clouds in afternoon	92	6	17980	24180	25.7

The power used in operating the test equipment, as determined from the electric meter readings, was so nearly alike in the two houses that no correction has been made for it. This would be expected as the equipment in the two houses was identical and most of the power load was continuous during the tests. The actual power used per test day was from 0.5 to about 1.0 kwhr, or from 5 to 10 per cent of the total cooling load. Hence the saving due to the heat-absorbing glass, expressed as a percentage of the total cooling load due to sunlight and atmospheric temperature only, was actually slightly higher than the values quoted in these tables, as these percentages are based on the total

TABLE 3. TEST ON EFFECT OF VENTILATING SLOTS

No. 1 House — Ordinary Glass + Heat-absorbing Heat-resisting Glass
No. 2 House — Ordinary Glass + Ordinary Glass

DATE—WEATHER	OUTSIDE TEMP.			INSIDE TEMP.				TIME	HEAT WITHDRAWN		SAVING DUE TO HEAT-ABSORBING GLASS %
				NO. 1 HOUSE		NO. 2 HOUSE					
	Beginning F	End F	Max. F	Min. F	Max. F	Min. F	Max. F		No. 1 House Btu	No. 2 House Btu	
8/21 Sunshine — Ideal.....	72.5	74.5	76.5	68.5	70	70	71	[a] 1-2 p.m.	2210	3180	29.8
								[b] 2-3	2760	3680	25.0
								[a] 3-4	2880	4360	33.9
								[b] 4-5	3070	4090	24.9
								TOTAL	10920	15280	28.5

[a] Ventilating slots in outside windows of No. 1 house opened.
[b] Ventilating slots in outside windows of No. 1 house closed.
Ventilating slots in No. 2 house left open during the entire test period.

heat removed from the test houses which included the heat generated by the power used in the testing equipment.

The second set of tests was made with single windows only, the one house having ordinary glass and the other the heat-absorbing glass windows. As in the previous set of experiments, check tests were made with the house and window combinations interchanged. The results are given in Table 4.

TABLE 4. RESULTS OF CHECK TESTS WITH SINGLE WINDOWS

DATE—WEATHER	MAX. OUTSIDE TEMP. F	TIME OF OPERATION HOURS	HEAT ENERGY REMOVED		SAVING DUE TO HEAT-ABSORBING GLASS %
			No. 1 House Btu	No. 2 House Btu	
No. 1 House — Single Windows — Heat-absorbing Heat-resisting Glass *No. 2 House — Single Windows — Ordinary Glass*					
9/1 Good — Few cirrus clouds.....	75	6	18060	21470	15.9
9/3 Clear in morning — rather dense cirrus clouds at noon — later clearing.....	79	6	23240	29010	19.9
9/5 Numerous cumulus clouds.....	76.5	6	15120	19290	21.6
9/6 Hazy in morning — cumulus clouds in afternoon.....	95.5	6	20410	25440	19.8
No. 1 House — Single Windows — Ordinary Glass *No. 2 House — Single Windows — Heat-absorbing Heat-resisting Glass*					
9/28 Clear except for few cirrus clouds..	69	5	16090	12220	24.0
10/8 Clear.....	76	6	29200	22970	21.3

Reviewing the results of the room cooling tests, it is very apparent that there were very considerable variations in the sunlight energy on the different test days. The total cooling load of course included conduction of heat through the windows and walls as well as the solar radiation. In the July tests when the outdoor temperatures were in general high, at the same time the high elevation of the sun reduced the effective window area, the solar radiation load due to the windows was a much smaller percentage of the total load. Therefore, the saving due to the heat-absorbing glass was a much smaller percentage of the total heat withdrawn than on days later in the season with lower outdoor temperatures and lower sun elevation. This is readily apparent from a comparison of the results in Table 2 with the values for the open slot arrangement in Table 3. The same explanation holds for the fact that the percentage saving

TABLE 5. TESTS TO DETERMINE EFFECT OF HEAT-ABSORBING GLASS IN REDUCING MAXIMUM TEST ROOM TEMPERATURES

No. 1 House — Double Windows — Ordinary + Ordinary Glass
No. 2 House — Double Windows — Ordinary + Heat-absorbing Heat-resisting Glass

DATE	WEATHER AT 9 A.M.	MAX. OUTDOOR TEMP.	MAX. ROOM TEMP.		DIFFERENCE DUE TO HEAT-ABSORBING GLASS
			No. 1 House	No. 2 House	
5/ 9	Sunshine with some clouds and haze.....	80 F	90 F	85 F	5 F
5/10	Semi-cloudy.....	89	94	90	4
5/11	Sunshine with horizon haze.....	75	89	80	9
5/12	Sunshine with horizon haze and some clouds	69	91	81	10
5/13	Sunshine with heavy horizon haze.....	73	83	77	6
5/14	Sunshine with horizon haze and some clouds	70	95	85	10
5/20	Sunshine with horizon haze and some clouds	96	110	102	8
5/28	Sunshine with horizon haze.....	85	100	94	6
5/31	Sunshine with horizon haze.....	94	110	104	6
6/ 1	Sunshine with horizon haze and some clouds	101	109	104	5
6/ 2	Sunshine with horizon haze and some clouds	100	113	108	5
6/ 4	Sunshine with horizon haze and some clouds	92	107	93	14

Average Difference 7.3 F

for single windows in Table 4 was in some instances actually greater than for the double window combination in Table 2.

An estimate of the saving in window load was made in a previous paper [4] based on the transmission data of various heat-absorbing heat-resisting glasses used singly and in combination with ordinary window glass in the inside sash. The estimated percentage saving for heat-absorbing 3 mm plate, which is practically identical with that used in the present tests, was 23.3 per cent with the single windows and 37.7 per cent in the case of the double window combination. The maximum values observed, as shown in Tables 2, 3 and 4, are 24.0 per cent for the single windows and 28.5 per cent for the double combination, and in the latter case the value is somewhat reduced because the ventilating slots were closed for part of the time. The agreement between the theoretical and observed

[4] Loc. Cit. See Note 3.

values is reasonably satisfactory, considering the uncertainties due to weather conditions and variations in the percentage of reflected energy due to changes in the angle of the incident light on the windows.

Before the refrigerating equipment was installed, a series of tests was made to determine the effect of heat-absorbing glass alone in reducing the maximum temperature attained in the test room. In these experiments both houses were equipped with double windows, No. 1 house having ordinary glass in both sash and No. 2 house having ordinary glass in the inside and heat-absorbing heat-resisting glass in the outside sash. The ventilating slots in all the outside sash were opened to permit as much ventilation as possible between the windows. There were no fans used in the houses and the maximum temperatures were obtained from maximum-minimum thermometers which were read each morning for the previous day and then reset. The results are given in Table 5.

The differences in maximum temperature attained in the two houses ranged from 4 to 14 F with an average of 7.3 F. These differences of course depend on the duration and intensity of the sunshine and, while on several of the test days the outdoor temperatures did not become at all uncomfortable, nevertheless the results show a very appreciable difference in favor of the heat-absorbing glass which would be of real value, for example, in cases where systems of night air cooling were used.

The effectiveness of heat-absorbing glass in increasing the comfort of an individual sitting in a window where he is exposed to direct sunlight is indicated by the results of temperature measurements taken during the period when the houses were cooled to 70 F. Maximum-minimum thermometers with blackened bulbs were suspended just inside the inner window so as to be exposed to direct sunlight. The maximum temperatures attained by these thermometers during the test runs with the various window combinations are given in Table 6.

These data show that the use of heat-absorbing glass would make a very appreciable difference to an individual sitting in a window exposed to sunlight. Another characteristic of heat-absorbing glasses is the faint blue or blue-green color tint. The color in the heat-absorbing glass under test was so slight that it was not noticeable after it was installed unless called to the attention of the observer, but it has a very pleasing effect on the eyes due to reduction in light glare. It is believed that a somewhat darker shade with consequent higher heat-absorbing efficiency would be acceptable as far as color is concerned and quite desirable from the standpoint of reduction of glare.

FUEL SAVING

The question of the effect of heat-absorbing glass on fuel cost in winter time is one of general interest. If this glass is effective in screening out the sun's heat in the summer time when it is undesirable, it would appear that it should be equally effective in winter weather when the sun's heat is desirable. The test houses were used to obtain some information on this point during the period from February 14 to March 28, 1934. Electric heating elements of 1500 w capacity were installed in each house and connected through a thermostatic relay which controlled the room temperature at 70 F. A fan was used to circulate the air in the room and was so placed that a constant current of air passed over the bimetallic element of the thermostat. The entire power used in each house was recorded with an electric meter. Using double windows

with ordinary glass in both sash on both houses, a test run was made to determine whether or not there was any difference in the heat insulation of the two houses. The correction was applied to the results obtained with various window combinations. Table 7 gives a summary of the tests with double window combinations.

The corrected difference of 8.8 per cent increase in heat required due to the heat-absorbing glass is probably somewhat higher than would be found for a complete winter period, as the test period of 10 days in which this was obtained included 5 entirely clear days and only 2 days in which the sun did not appear

TABLE 6. TEMPERATURES INSIDE WINDOWS IN SUNSHINE WITH ROOM TEMPERATURES CONTROLLED AT 70 F

No 1 House		Max. Window Temp. F	No. 2 House		Difference in Max. Window Temp. F
Date	Window Arrangement		Window Arrangement	Temp. F	
7/16	Ordinary + Ordinary...	99	Ordinary + Heat-absorbing Heat-resisting Glass	83.5	15.5
7/17	" ...	92	" ...	81	11
7/19	" ...	98	" ...	85	13
7/20	" ...	96	" ...	83	13
7/22	Ordinary + Heat absorbing Heat-resisting Glass	88.5	Ordinary + Ordinary...	100	11.5
7/23	" ...	85.5	" ...	96	10.5
7/25	" ...	90	" ...	101	11
7/30	" ...	85	" ...	94.5	9.5
8/21	" ...	89.5	" ...	100.5	11
9/ 1	Single Heat-absorbing Heat-resisting Glass...	95	Single Ordinary.....	100	5
9/ 3	" ...	95	" "	103	8
9/ 5	" ...	98.5	" "	103.5	5
9/ 6	"	95.5	" "	101.5	6
9/28	Single Ordinary.....	107.5	Single Heat-absorbing Heat-resisting Glass...	91	16.5
10/ 8	" "	105	" "	95	10

Average DIFFERENCE 10.4 F

at some time during the day. The comparatively small difference in the heat required with the heat-absorbing glazing in the outer sash is accounted for by the fact that the hours of sunshine per day are comparatively few during the cold season in Corning.

Another test run was made to show the heat saving through the use of storm sash glazed with heat-absorbing glass as compared with single sash containing ordinary glass. The results are given in Table 8.

Thus in cases where no storm sash are now used, the addition of an outer sash with a heat-absorbing glass for air-conditioning purposes will result in a very appreciable saving in the quantity of heat required in cold weather.

TABLE 7. SUMMARY OF TESTS WITH DOUBLE WINDOWS

HOUSE NO.	WINDOW ARRANGEMENT	TEST PERIOD	OUTSIDE TEMP.		HEAT REQUIRED KWHR	DIFFERENCE %
			Min. F	Max. F		
1	Double — Ordinary.....	3/18 to 3/28	15	63	189.0	
2	" "	"	15	63	192.6	
	Correction to be applied to No. 2 House = 1.9%					
1	Double — Ordinary.....	2/14 to 2/24	−5	46	225.6	
2	Double — Ordinary + Heat-absorbing Heat-resisting Glass.....	"	"	"	250.6	
2	Double — Ordinary + Heat-absorbing Heat-resisting Glass.....	"	"	"	245.7 (corrected)	8.8 (Increase due to heat - absorbing heat - resisting glass)

EFFECT OF SUN AND SHADOW

In connection with the use of heat-absorbing glass there naturally arises the question of the chemical durability of the glass, the permanency of the heat-absorbing effects, and the ability of the glass to withstand the service conditions that may be encountered. As far as chemical durability is concerned, the boro-silicate heat-resisting type of glass is believed to be quite equal and may even be superior to ordinary window glass. The heat-absorbing properties are obtained by means of the chemical composition and appear to be as permanent as

TABLE 8. TEST RESULTS OF STORM SASH WITH HEAT-ABSORBING GLASS AND SINGLE SASH WITH ORDINARY GLASS

HOUSE NO.	WINDOW ARRANGEMENT	TEST PERIOD	OUTSIDE TEMP.		HEAT REQUIRED KWHR	PERCENTAGE DIFFERENCE %
			Min. F	Max. F		
1	Single — Ordinary.....	2/24 to 3/5	−9	69	235.6	
2	Double — Ordinary + Heat-absorbing, Heat-resisting glass	"	"	"	193.2	
2	Double — Ordinary + Heat-absorbing, Heat-resisting glass	"	"	"	189.5 (corrected)	19.6 (Saving due to addition of heat - absorbing double glazing)

the glass itself. As far as service conditions are concerned, the chief requirement is that the glass should have the necessary thermal endurance so that breakage will not occur when shadows are cast across the window or a cold rainstorm suddenly chills the glass. Through the courtesy of Dr. H. M. Cook, of West Texas State Teachers' College, some tests were made at Canyon, Texas, on a set of 4 different types of heat-absorbing glass, 3 of them being of the non-heat-resisting type and one borosilicate heat-resisting. The test windows were 24 x 36 in. in size and approximately ¼ in. in thickness. They were puttied in the usual manner into frames which formed the covers of box-like structures about 6 in. deep, the back being of gray insulating board to simulate conditions

TABLE 9. RESULTS OF TESTS ON 4 TYPES OF HEAT-ABSORBING GLASS

DATE	TIME	SHADE TEMPERATURE	SUN TEMPERATURE	REMARKS
8/ 2	2:00 p.m.	99.5 F	115 F	
8/ 3	"	97.5	111	
8/ 4	"	97.5	113	
8/ 5	"	104	115	
8/ 6	"	102	124	No. 3 checked
8/ 7	"	99.5	116.5	
8/ 8	"	102	118.5	No. 3 checked further under shaded area
8/ 9	"	100.5	116.5	
8/11	"	104	120	No. 3 checked further
8/12	"	106	131	No. 4 checked under shadow
8/13	"	102	124	
8/14	"	104	122	
8/15	"	105	116.5	
8/16	"	106	125.5	Nos. 1, 3, 4 checked
8/17	"	106	116.5	
8/18	"	105	116.5	
8/19	"	102	116.5	
8/20	"	104	111	
8/21	"	102	125.5	

No. 2 (heat-resisting, heat-absorbing glass) unbroken at conclusion of test.

that would exist in a building. These boxes were mounted so that the windows were normal to the sun's rays and in each case a shadow was cast across the glass by means of a board nailed across the window frame. The results are given in Table 9.

The shade temperatures were undoubtedly high at the time the test was made, but the breakage was caused by the difference in temperature between the glass in the shadow and the sunlight rather than by the high atmospheric temperatures. Hence, breakage will undoubtedly occur whenever a combination of intense sunlight and shadow results in thermal stresses exceeding the thermal endurance of the sheet in question. The heat-resisting type of heat-absorbing glass with a thermal expansion coefficient slightly greater than one-half that of

the usual heat-absorbing glass possesses a thermal endurance almost twice as great, giving an increased safety factor and permitting much larger sizes of heat-absorbing plate to be used.

SUMMARY

(1) A reduction in the cooling load varying from 9 to 28.5 per cent was observed in a test house equipped with heat-absorbing heat-resisting glass used as an outside window with ordinary glass in the inside sash and from 16 to 24 per cent with single heat-absorbing glazing. The combination of double windows is theoretically much more effective than single glazing and the explanation of the above overlapping ranges of values lies in variations due to weather conditions and the seasonal change in the elevation of the sun.

(2) With no cooling equipment in operation the test house with the heat-absorbing double window combination averaged 7.3 F lower maximum temperature on sunny days than a similar test house with ordinary glass in both outside and inside sash. With air-conditioning equipment in operation, maximum recording thermometers with blackened bulbs, suspended inside the windows in the sunshine, averaged 10.4 F lower when heat-absorbing glass was used.

(3) Heat required in a 10-day period in a winter test was approximately .9 per cent more in a test house having double windows with heat-absorbing glass in the outside sash as compared with a similar house having ordinary window glass in the outside sash. A test house having double windows with heat-absorbing glass in the outside sash required approximately 20 per cent less heat than a similar house having single windows of ordinary glass.

(4) Heat-absorbing borosilicate glasses appear to have a satisfactory chemical durability and permanency of physical properties. Owing to the heating of the glass in sunlight due to the high absorption of energy, the thermal endurance of heat-absorbing glass becomes of considerable importance. The borosilicate type of glass offers a greater factor of safety owing to its low thermal expansion coefficient.

The author wishes to express his gratitude for the co-operation of various members of the Corning Glass Works Laboratory staff and their assistance in collecting the data and calculating the results, and particularly to thank Dr. Cook, Canyon, Texas, for his part of the test work.

DISCUSSION

S. R. LEWIS: This paper brings to our attention the very potent influence of sunshine in winter.

It has long been the custom to allow for extra heating on the north sides of buildings in this latitude, presumably because of the cold winds from the north. I think we are coming to realize that the provision of extra heat on the north is not so much due to this effect as to the absence of sun heat, and it seems apparent that temperature control within a building having some rooms with exclusively southern exposure and some rooms with exclusively northern exposure is the important factor.

In winter the sunlight penetrates a much greater distance into a south room and warms the floor and the objects which it touches than in summer. I believe that the study which is being given to sun heat in the design of cooling systems will bring about improvement in the design of heating systems.

No. 1019

COOLING AND AIR CONDITIONING IN AN ALE BREWERY

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MONTREAL, QUE., CAN.

THE purpose of this paper is to describe the methods adopted in the design of the cooling and air-conditioning equipment for the fermenting room of an ale brewery and also to describe certain constructional features of the installation, which it is believed will be of interest. Ventilating and air conditioning problems in other parts of the plant will be touched on with the idea of indicating their scope, rather than with any intention of giving complete descriptions or discussion.

The fermenting room of an ale brewery is undoubtedly the department in which an adequate cooling and ventilating system is most important. Other departments such as the wort cooler room and storage rooms also offer their problems but these are of minor importance when compared to the fermenting room. This paper will treat principally of the fermenting rooms of Molson's Brewery, Montreal, the methods of conditioning of which it is believed offer several points of interest.

It has been found in breweries with which the authors are familiar that even temperature and humidity in the fermenting room tend to greater uniformity of product both as to quality, flavour, and rate of fermentation. The clean filtered air lessens the danger of contamination of the brew by air borne wild yeast cultures, mould spores, or other organisms, while the dryer air of the room lessens the condensation on walls and other apparatus with consequent reduction in mould growths. The lessening of condensation on walls, etc., also permits a considerable saving in maintenance of the building due to the greater life of paint work and reduced necessity of frequent scrubbing and disinfecting.

In the winter, when heating is required with consequent necessity of added humidity in the air, the drying-out of empty tuns is retarded and yeast remains in a much healthier condition.

There are 5 fermenting rooms in the brewery being considered, 4 of which are air-conditioned. These rooms are of the following sizes:—

Room *A*..... 44 ft × 36 ft × 18 ft
Room *B*.....175 " × 37 " × 18 "

* McDougall & Friedman, Consulting Engrs., Montreal, Can., and New York, N. Y.
** McDougall & Friedman, Consulting Engrs., Montreal, Can., and New York, N. Y.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June 1935, by C. P. Creighton.

Room *C*.....122 ft × 38 ft × 18 ft
Room *D*.....122 " × 38 " × 18 "

Each room contains 2 rows of fermenting tuns, 1 row along each wall. The rooms are partially divided into 2 levels by a gallery 9 ft from the floor through which the tuns project. This gallery has an opening 8 ft wide in the center and extending the entire length of the room. Fig. 1 shows a typical cross section of these rooms.

The heat loss in winter and heat gain in summer, due to outside conditions, were calculated in the conventional manner, the only rather unusual feature being the very large number of factors for heat transmission through building

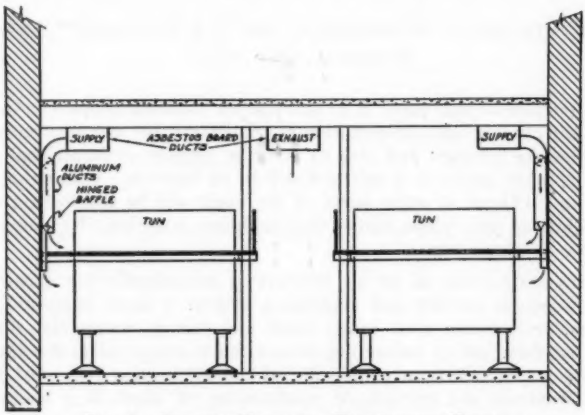

FIG. 1. SECTION THROUGH FERMENTING ROOM

construction owing to the number of different types of walls and their varying ages and conditions. Parts of the buildings are over 100 years old and numerous alterations and additions have been made since the original construction, so that it was necessary in most cases to drill the walls to determine their character and condition.

Wall transmission factors were calculated according to the methods recommended in THE A. S. H. V. E. GUIDE and, while the calculations were somewhat laborious, due to the large number of types of construction used in different parts of the walls, floors and ceilings, and also to the very different temperatures obtaining in adjacent rooms or the outside at different seasons of the year, the method involved no departure from standard practice. Separate calculations were made to determine heat losses or gains in both winter and summer. In one room particularly—Room *A*—these quantities practically balanced at all times since the heat lost or gained through outside walls or from adjacent warm rooms was almost equal in most cases to that lost to adjacent cold rooms. The rooms were all well insulated with corkboard and were provided with gasketted refrigerator doors and triple glass windows.

The Btu losses or gains per hour due to wall transmission, but not including ventilation or process requirements, were:—

	HEAT LOSS, WINTER	HEAT GAIN, SUMMER
Room *B*	48,000 Btu	68,600 Btu
Room *C*	23,800 Btu	10,229 Btu
Room *D*	39,000 Btu	20,900 Btu

The rooms were all figured for the same inside and outside conditions. Inside temperatures were to be maintained at 55 F with a relative humidity of 70 per cent to 75 per cent. Outdoor conditions were assumed as—20 F in winter, and 90 F and a relative humidity of 70 per cent in summer. The design of the whole plant made provision for the introduction at all seasons of a maximum of 2500 cu ft of outside air per minute for ventilation. The fans and duct work were designed to recirculate 28,000 cu ft of air per minute.

The indoor conditions of an ale brewery are somewhat different from those usually maintained in lager beer breweries where fermenting room temperatures are usually around 40 F and indoor humidity can also be kept lower owing to the general use of closed steel fermenters. Ventilating requirements of such plants are also reduced as the carbon dioxide gas produced in the process is conducted by pipes to a point outside the fermenting room.

The heating and cooling ventilating requirements to take care of conditions due to the process of fermentation were more difficult to determine, the data available being rather meagre and the conditions varying in each individual plant due to differences in the methods of fermentation used.

During the process of fermentation, sugars in the wort or unfermented ale are acted on by the yeast and converted to alcohols and carbon dioxide gas with consequent production of heat. Most of this heat is carried off by cold water circulating in copper coils inside the fermenting tuns. There is, however, a certain amount of heat and moisture thrown off from the surface of the fermenting ale. There is also a large amount of carbon dioxide gas produced, this gas flowing over the edge of the tuns into the room. The rate of fermentation is controlled throughout the process by varying the temperature so that heat and gas are produced at a varying rate at different periods throughout each brew. The rate of production of heat was calculated from the known rate of change of sugar into alcohol and gas and the amount of heat carried away by the cooling water measured and deducted. The difference was assumed to be the heat lost to the air of the room either by convection or the evaporation of moisture. Actual figures are not quoted for this calculation, as they involve private information as to temperatures and rates of fermentation which have an important bearing on the quality and flavour of the ale and which are in the nature of trade secrets for each plant where such a system might be installed.

Owing to the scrupulous cleanliness in these rooms, a large amount of washing is done involving a large moisture pick-up from wet walls, floors and tuns. This amount could only be estimated from experience in similar installations and would have to be estimated with a view to actual conditions.

The production of carbon dioxide gas during fermentation was calculated in a manner similar to that indicated for the production of heat and was also built up step by step for successive brews. Owing to the steady orderly production of the gas, the concentration near the floor was found to be very high, sometimes reaching 5 per cent to 6 per cent at heights of 6 in. or less, although decreasing 2 per cent to 3 per cent at the breathing line. By taking advantage

of this condition and exhausting air from the rooms at very low levels while recirculating air from a point near the ceiling, the amount of air from outside

FIG. 3. REAR OF DEHUMIDIFIER SHOWING AMMONIA CONNECTIONS

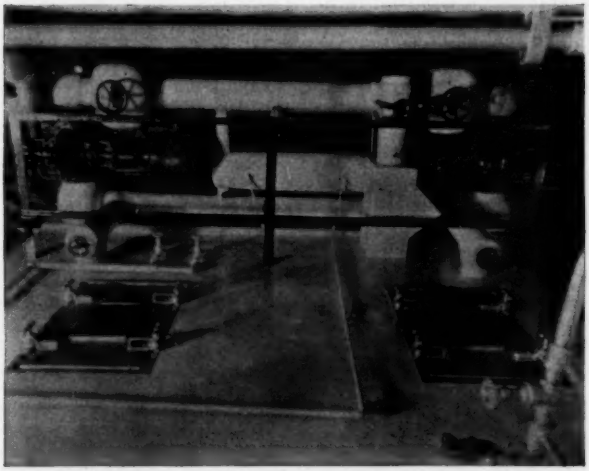

FIG. 2. FRONT OF DEHUMIDIFIER LOOKING DOWN

necessary for ventilation was greatly reduced with consequent saving of heat and refrigeration.

After considering the various methods of cooling, such as unit coolers, three central plants, or a single central plant, the latter was adopted.

The conditioning equipment consists of duplicate circulating fans; dehumidifier unit with by-pass ammonia cooling coils, water sprays, and duplicate water

circulating pumps; preheater in fresh air inlet, filter in fresh air inlet, reheaters in supplies to each room, and automatic control thermostats, dampers and diaphragm valves. All this apparatus is located in a suction chamber to which all return ducts are brought. The suction chamber is brilliantly lighted and has large glass windows for observation and display, all apparatus being finished in white enamel. Figs. 2, 3 and 4 show general views of apparatus in this room.

Temperature of air delivered from the dehumidifier is controlled by a thermostat in the return duct from Room *B*, this room being the one requiring the greatest temperature differential between supply and return when cooling, and the least when heating. Individual reheaters in the ducts supplying *C* and *D* are controlled by thermostats on the return ducts from these rooms. The re-

FIG. 4. DUCTS AND HEATERS IN TOP OF APPARATUS ROOM

heaters also serve as heaters for the rooms in the winter. Fig. 9 shows a diagram of the dehumidifier, fans and heaters.

No separate fresh air fan is provided, the duct being connected to the inlet of the dehumidifier and the quantity being kept constant by a velocitystat located in the inlet and controlling the suction on the dehumidifier. Dry type filters are used to clean this air before entering the sprays.

The duct work presents several unusual features. Owing to the fact that the system had to be installed in an existing building, it was impossible to provide for the concealment of ducts. It was, therefore, necessary to run ducts on the ceilings of rooms over the top of fermenting tuns. It was considered that there was danger under certain conditions of ducts sweating and dripping into these open tuns with the possibility of contamination of the product which might possibly ruin a brew and as the raw material is of considerable value and as such contamination might also cause great inconvenience, it was decided to

build all such exposed ducts of insulating material. A composite material consisting of 2¼-in. layers of hard asbestos board with ½ in. of wood fibre board between was chosen. These sheets were fastened together with galvanized angles, strips, and bolts as shown in Fig. 5. All main ducts, vertical and horizontal were made large enough for a man to crawl through for cleaning. Vertical supply drops exposed in the room were constructed of polished #14 gage aluminum. Large vertical ducts were fitted with permanent steel ladders inside to aid in cleaning and vertical supply drops arranged so that a long-handle brush could be pushed up from the bottom. Air distribution was controlled by numerous volume dampers rather than by duct sizes as the dimensions necessary for cleaning were often much larger than necessary for the quantity of air.

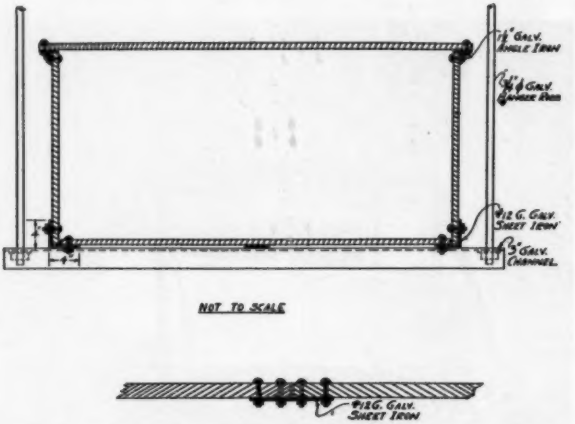

FIG. 5. DETAIL OF FIBRE DUCT CONSTRUCTION AND OF CROSS JOINTS

In addition, duct sizes were chosen so that standard sheets of the insulating board could be cut without waste. Figs. 6, 7 and 8 show general and detail views of the duct work. The cost of this duct work was about double that of sheet metal, though little if any greater than the cost of metal duct covered with insulating material.

Other departments of the brewery, principally the hot wort cooler room, storage cellars and bottling shops, offer a field for air conditioning with or without cooling.

In the hot wort cooler room it is the usual practice to run the hot liquid over the pipes of a Baudelot type cooler. This serves the dual purpose of cooling and aerating the wort. Since the liquid is at boiling temperature when it flows on to the cooler, a large amount of vapor is released necessitating powerful supply and exhaust fans for its removal. The supply air should be filtered to prevent contamination of the wort by dust, wild yeast or bacteria. Care must also be taken to keep only a very slight plenum in the room in order to avoid driving moisture into the building construction. Heaters for this air, preferably

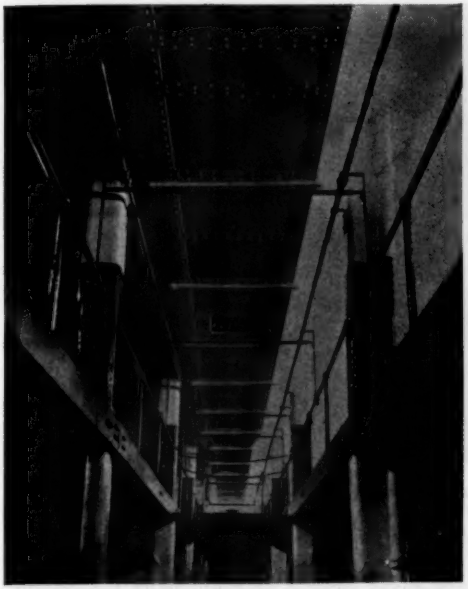

FIG. 6. MAIN EXHAUST TRUNK

thermostatically controlled, are desirable in order to reduce fog in the room and to dry up condensation on the walls.

The storage cellars and hop storages also offer a field, though less extensive, for air conditioning. The reasons for this are very similar to those already cited for the fermenting rooms, namely uniformity of conditions for aging and storage, prevention of sweating of walls and apparatus with consequent greater cleanliness and ease of maintenance. Bottle shops and other such departments present only conventional ventilation problems of heating in winter, and the removal of heat and vapor in summer.

The authors regret that it is impossible to present accurate operating data concerning the fermenting room installation described, as complete experience extends over the winter months only and there has been no weather as yet suitable for a complete test

FIG. 7. MAIN SUPPLY TRUNK SHOWING ALUMINUM SUPPLY DROPS

FIG. 8. DETAIL OF ASBESTOS BOARD DUCT ELBOW

on the cooling operation. The authors hope, however, to present such operating data to the Society at some future date. Winter operation results were very satisfactory, the apparatus being easily capable of maintaining the designed inside temperature and relative humidity as low as 55 per cent if desired. Heat losses checked by the quantity and temperature of air delivered and recirculated from the rooms checked very closely with the design figures.

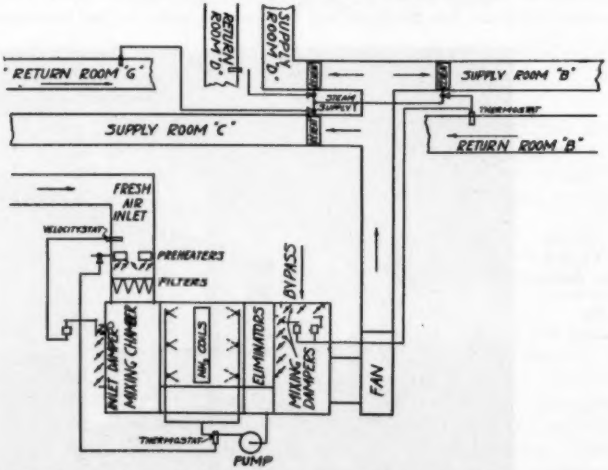

FIG. 9. DIAGRAM OF APPARATUS AND CONTROLS IN RECIRCULATED AIR CHAMBER

No. 1020

AN ALL ELECTRIC HEATING, COOLING AND AIR CONDITIONING SYSTEM

Installation and Performance of Plant Using the Heat Pump Principle

By PHILIP SPORN * (*NON-MEMBER*), NEW YORK, N. Y., AND D. W. MCLENEGAN ** (*MEMBER*), SCHENECTADY, N. Y.

INTRODUCTION

THE use of a refrigerating machine as a heat pump for the heating of a house in winter is not new, having been first suggested more than 80 years ago by Lord Kelvin.[1] Others [2, 3] have pointed out that at least in southern climates refrigerating machines installed for the purpose of cooling houses in the summer, might, by the use of the above principle, also be installed for the heating of houses in the winter-time. In recent years a number of heat pump installations have been built and described [4, 5] and at least one installation made embodying the idea of using a single set of refrigeration equipment to carry out the heating and cooling with the same equipment.[6]

As is well known, the ideal Carnot efficiency of a reverse refrigeration cycle operating as a heat pump is given by the equation $E = \frac{T_1}{T_1 - T_2}$. Under this arrangement a heat pump taking heat from a source at a temperature of 60 F, and discharging heat at a temperature of 135 F, would have an ideal efficiency of 7.95. With the outside temperature at —20 deg, the ideal efficiency drops

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[1] Thomson, William (Lord Kelvin), On the Economy of Heating and Cooling of Buildings by Means of Currents of Air, *Glasgow Phil. Soc. Proc.*, v. 3, December, 1852.

[2] Stevenson, A. R. Jr., Cooling Homes, A Field for Refrigeration, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926.

[3] Stevenson, A. R. Jr., Refrigeration, *Journal of the Franklin Institute*, Vol. 208, No. 2, Aug., 1929, p. 143 and following.

[4] Haldane, T. G. N., The Heat Pump—An Economical Method of Producing Low-grade Heat from Electricity, *Electric Review*, v. 105, pp. 1161-1162, December 27, 1929, and *I. E. E. Journal*, v. 68, pp. 666-675, June, 1930.

[5] Wilkes, Gilbert and Marbury, R. E., House Heated by Pump with 5 to 1 Pick-up Ratio, *Electrical World*, Vol. 100, December 17, 1932, p. 828 and following.

[6] Doolittle, H. L., Edison Building Heated and Cooled by Electricity, *Power*, v. 74, pp. 348-351, September 8, 1931.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June, 1935, by D. W. McLenegan.

to 3.74. With the actual efficiency running in the order of 60 per cent of the ideal, it is apparent that under climatic conditions where temperatures of the order of zero to —20 deg have to be encountered, the efficiency of a heat pump drops very rapidly and may approach a value close to unity. The net effect of this is one of two things:

(a) If the heat pump is selected with a view of being adequate under the extreme low temperatures, then a size of compressor is involved that is so much larger than that necessary for refrigeration, that it is not only uneconomical from the standpoint of equipment investment cost, but is also out of balance from the standpoint of having approximately the same equipment performing the job in summer and in winter; or

(b) If an attempt is made to avoid that condition, then some provision has to be made for taking care of the extreme cold weather conditions by auxiliary means. These may take the form of fuel heaters or direct electric heaters, and some of the installations [6] have resorted to these methods.

In adopting that solution, however, there is definitely raised the question of whether the heat pump as a method of heating is a self-reliant piece of equipment or could even be developed as such. The particular installations previously described obviously made no attempt to answer these questions or to solve the problem of economical balance between heating and cooling equipment. In brief, none of the heat pump installations previously made have been complete, or all heat pump installations.

It has been apparent for some time to those responsible for the development described in this paper, that if electric heating of homes, stores, offices and similar buildings is to overcome the handicap of, first, a 70 per cent heat loss in going from thermal to electrical energy, involved even in the best steam generating plant, and, second, even more important, the excessive costs of large distribution facilities necessary in any straight 1:1 heat conversion process when going from electrical to thermal energy; and if electric heating is to overcome the handicap of the fixed charges involved in cost of equipment, then the reverse refrigeration cycle of heating, which makes possible the use of the same equipment for cooling in summer and heating in the winter, and which gives the maximum economy in use of electrical energy, with a minimum requirement of electrical capacity to take care of a given installation, would be able to overcome all such handicaps. Hence, in considering the heating, cooling and air conditioning system for the particular building described in this paper, the first decision made was that it would be an all electric and all compressor system. Having decided that point, and in order to meet the difficulties with regard to unbalance between summer and winter refrigerator capacity already pointed out, and the difficulties of the concomitant heavy investment cost that would be involved in an all compressor system if the source of heat were the outside atmosphere (since the building located in southern New Jersey had cycles of zero F and even lower temperatures to contend with) two possible solutions were considered:

(1) A storage system. With this system heat obtained by the heat pump from the outside air under favorable (mild temperature) conditions would be stored at a moderate temperature (between 60 deg and 80 F) using that portion of the compressor capacity not required for immediate heating. This storage would be made available as a source of heat for the compressors under conditions of exceedingly low

outside temperatures when conditions for obtaining heat directly by the heat pump from the outside were unfavorable, and

(2) A more simple system, in which the heat would be obtained from water supplied at a constant temperature from a deep-well.

Both of these solutions were thoroughly explored with the definite knowledge that at least in northern climates the first of them would be more general and of greater applicability than the second. However, after considering the various technical problems involved which had theretofore not been definitely worked out, it was decided not to attempt too much in the first installation and to try the simpler of the two solutions, that is, utilize a deep-well water supply as the source of heat in the winter and as a sump for the rejection of heat in the summer time.

THE BUILDING

The building heated, cooled and air conditioned by the equipment described

FIG. 1. SALEM OFFICE BUILDING

in this paper is a two-story and basement, modern but very simple, brick and steel reinforced concrete structure belonging to the Atlantic City Electric Co., a subsidiary of the American Gas and Electric Co., and located at Salem, New Jersey. The lower floor is devoted to a display room, service department and offices, while the entire second floor is used as a combined demonstration hall and auditorium. The two photographs reproduced as Figs. 1 and 2 show clearly the character of the building. The latter, too, shows very distinctly the arrangement of grilles at the base of the walls for exhausting the cold air and the registers along the ceiling for delivering the conditioned air to the room. The basement of the building is used for storage and contains the pump used to give the water supply, and the air conditioning equipment. Fig. 3 shows the well and pump installation.

The brick walls and roof are insulated with 2 in. of cork. There are 1240 sq ft of windows. The total volume of the building is approximately 76,800 cu ft, of which the first and second floors represent a volume of 55,200 cu ft.

From the building plans, the heating requirement for 0 F outdoor temperature is given in the accompanying table.

FIG. 2. INTERIOR VIEW SHOWING INLET AND OUTLET GRILLES

	BTU/HOUR
Conduction loss, first and second floors	143,000
Warming the ventilation air for first and second floors	107,000
Basement heat loss	35,000
Total	285,000

The heat loss of the basement does not vary greatly with outdoor temperature, being affected principally by the ground temperature, which is constant.

For summer operation, it was planned to cool the first and second floors to 80 deg dry-bulb temperature with 50 per cent relative humidity, assuming outdoor conditions at 95 deg dry-bulb and 75 deg wet-bulb. Allowing for 1200 cfm of ventilation air, the maximum summer load is 108,000 Btu/hour distributed as shown in the accompanying tabulation.

FIG. 3. WELL AND WATER PUMP

SENSIBLE HEAT	BTU/HOUR	
Conduction	33,300	
Solar radiation	18,300	
Ventilation air	19,600	
40 people	8,800	
		80,000
LATENT HEAT		
Ventilation air	20,800	
40 people	7,200	28,000
Total		108,000

The total air flow was based on the following conditions:

(1) Air flow to be adequate for maximum winter heating at relatively low discharge temperature.

(2) Summer cooling to be accomplished with air not more than 15 deg below room temperature.

(3) Air flow to be small enough to avoid excessive duct sizes.

A total air flow of 5500 cfm was selected for conditioning the first and second stories of the building, which represent a volume of 55,200 cu ft. The basement is heated by the machine *losses*, but is not air conditioned. Thus, air is circulated through the first and second floor space at the rate of one change every 10 min, consistent with liberal practice. The system is designed for the admission of 1200 cfm of outdoor air, with provision for increasing this quantity to 1700 as desired. The average occupancy of the building is about 25 people. The ventilation air intake thus represents 48 cfm per person for summer or winter conditions, which is rather high. However, in terms of the maximum expected, which is 60, it represents a value of only 20 cfm per person and it was felt that until actual experience indicated that less ventilation air could be employed, it was not desirable to go below this value.

With 0 F outdoor temperature, the heat balance was planned as in Fig. 4. This diagram shows that the heat pump equipment alone is not quite adequate for a sustained period of zero weather, since the capacity was planned to be 260,000 Btu/hour and the maximum requirement was 285,000. It was felt, however, that the selection of this capacity was justified, since the building has a large lighting load which could be called on in extreme cases. Furthermore, the building is well insulated and would not require the maximum heating effect, except during a prolonged period of zero weather, which very seldom occurs.

THE SYSTEM

The system comprising the air conditioning installation is shown schematically in Fig. 5 and a description of its operation is given.

A deep-well pump with 3 hp, 3-phase motor, forces water through 4 water coolers in series. As refrigerant evaporators, these units are connected to 4 compressors and thence to the condensers (air heaters) as separate refrigerant circuits. Units Nos. 2 and 3 are arranged with reversible connections for summer operation, and for simplicity, these 2 machines are always operated as a single unit, since they are adequate to carry the summer cooling load.

Outdoor air entering the system passes through a preheating coil, not shown

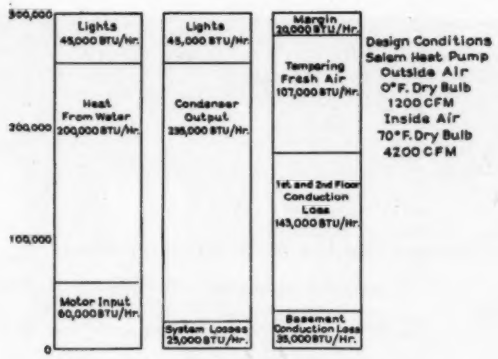


FIG. 4. HEAT BALANCE DIAGRAM

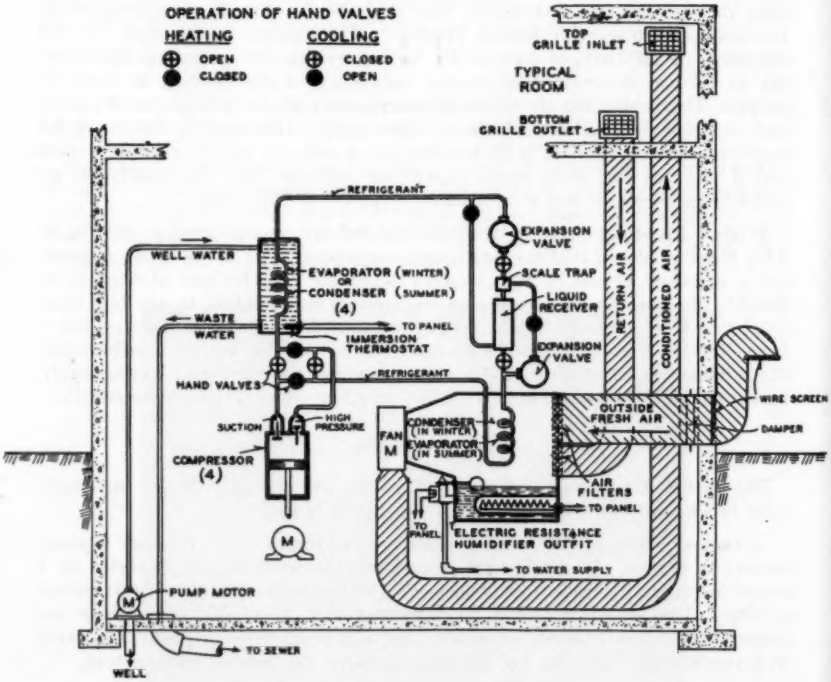


FIG. 5. SCHEMATIC LAYOUT OF HEATING SYSTEM

in the diagram of Fig. 5, which is the condenser of heat pump No. 1. This unit, controlled by a thermostat in the entering air, operates continuously when the outdoor temperature is below 40 deg. The ventilation air is then mixed with the recirculated air, and passes in succession over the condensers of Nos. 2 and 3 and No. 4 heat pumps, absorbing heat to a degree consistent with the heating requirements. An electrically heated humidifier adds moisture to the air, under control of a humidistat. The air is then distributed through the building through a conventional high velocity duct system, using overhead horizontal delivery and baseboard returns. The velocity in the supply mains is 1400 fpm, in the branches, approximately 1000, and at the outlets approximately 750, which permits adequate travel and mixing with the room air.

Since all air delivery, whether heated or cooled, enters through high wall registers, the air distribution had to be planned to avoid stratification during the heating season. This problem was successfully solved by locating a number of ample return openings at floor level, to remove the cooler air, and by using a relatively large total flow of air in proportion to the heat delivered. Under the most severe conditions, the temperature of the air delivered does not exceed 105 deg; and the air at this temperature is not too light to mix readily. With continuous circulation and a minimum cyclic variation of the air delivery temperature, it was expected that comfortable conditions would be maintained, as regards both the vertical temperature variation and the temperature-time variation. Under maximum load conditions, the refrigerant condensing temperature of the last heating unit (No. 4) was planned not to exceed 135 F.

Considering the heat pump only, greater efficiency could have been achieved by delivering air at a lower maximum temperature, since the compressors could then have been operated at lower condensing temperatures and hence with lower power input. This, however, would have required larger ducts, since a greater air flow would have been needed for the same heat delivery. Conversely, the use of a smaller quantity of hotter air would have impaired the heat pump performance, as well as the vertical temperature distribution. The authors believe that the outlet temperature selected represents a good compromise for this installation, although climate, building construction and further design and operating experience may call for a modification of this figure.

The compressors are 4 cylinder, single acting units, operating at 400 rpm, each belted to a 5 hp, 3-phase, 60 cycle, 1800 rpm induction motor, and are of a type commonly used in comfort cooling installations. Although the air-cooled condensers, which heat the conditioned air in winter, are not built integrally with the compressors, each compressor has its own liquid receiver, through which the refrigerant passes before returning to the evaporator.

The sizes of the air-cooled condensers have been proportioned to give efficient operation, taking into account also the first cost of this surface. The condenser for machine No. 1, which operates only as an air preheater in winter, consists of 4 coil sections and is relatively small, since the air entering through this condenser is cold and the unit operates at low discharge pressure. Machines Nos. 2 and 3 each have a 5-section condenser which the air enters at an average temperature near 60 deg in winter. Condenser No. 4 consists of 10 sections, since it must dissipate heat to air which has already been somewhat heated in cold weather by units Nos. 2 and 3; hence, this surface is made large to reduce

the refrigerant-to-air temperature difference, and thereby improve the coefficient of performance of this machine.

A general view of the physical arrangement and appearance of the equipment involved is shown in Fig. 6. A view of the 4 compressors is shown in Fig. 7. Fig. 8 shows a detailed view of the evaporators (water coolers) and

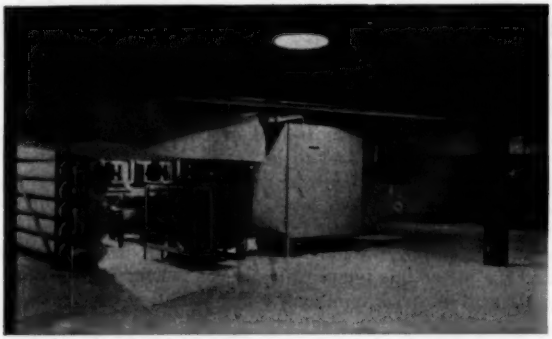


FIG. 6. CLOSED VIEW OF CONDENSER AND HUMIDIFIER (HEATING CYCLE)

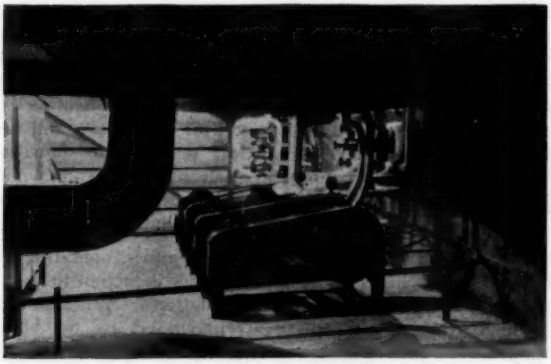


FIG. 7. COMPRESSOR INSTALLATION

Fig. 9 shows the condensers (air heaters) including the filtering and humidifying arrangement.

The operating sequence of machines to meet the heating demand is shown in Fig. 10. An indoor thermostat starts and stops the machines, but their availability is determined by separate thermostats, responsive to outdoor temperature and located in the incoming air stream. Thus, machine No. 4 runs alone and intermittently at outdoor temperatures above 54 deg. From 54 deg to 40 deg, machines 2 and 3 are also available. Below 40 deg machine No. 1 is added

and runs continuously. Below 23 deg machines Nos. 1 and 4 operate continuously and Nos. 2 and 3 are available.

The purpose of the sequencing scheme is to operate at all times with a minimum of excess capacity, and hence with lowest possible refrigerant condensing temperature. This has two advantages, namely:

(1) For any required average rate of heat output, continuous operation with low

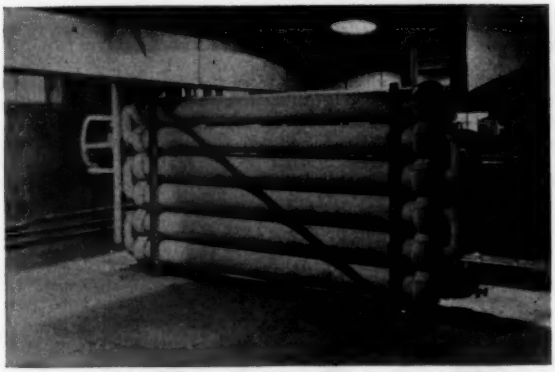


FIG. 8. EVAPORATORS (HEATING CYCLE)

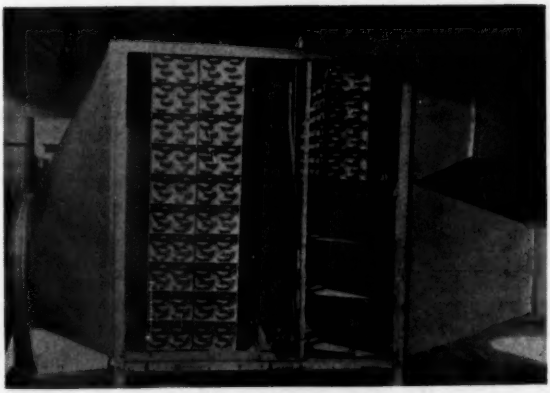


FIG. 9. OPEN VIEW OF CONDENSER AND HUMIDIFIER (HEATING CYCLE)

condenser pressure yields a higher coefficient of performance than intermittent operation at high condenser pressure and higher momentary output.

(2) It is desirable from the comfort standpoint to have continuous air delivery, using slightly heated air in mild weather, and increasing the air temperature in colder weather. In contrast, a single machine, or 4 machines always operated concurrently, would cause the hottest air to be delivered intermittently in the mildest weather.

The remainder of the control scheme is shown in Fig. 11 and centers about this selection of units. The air conditioner fan is operated continuously and at constant speed. By a system of interlocks, the water pump is started automatically whenever any of the compressors are started by the indoor thermostat. In case of pump failure, the water coolers are protected from freezing by immersion thermostats (with 40 deg setting), located in the lower sections of the water cooler. The humidifier in the air conditioner is started and stopped directly in response to a room humidistat.

The changeover from winter to summer operation involves:

(a) Transfer of the control circuit to reverse the operation of the room thermostat. For summer operation, a cooling rather than a heating requirement is indicated when the thermostat closes its contacts.

(b) Manipulation of valves. The air-cooled condensers now become refrigerant evaporators, in which the flow is regulated by automatic expansion valves. These valves, normally by-passed in winter, are cut in by switching to a different refrig-

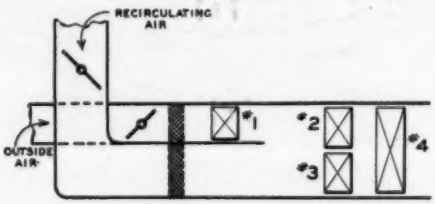


FIG. 10. ARRANGEMENT OF AIR COOLED CONDENSERS FOR WINTER HEATING

erant manifold connection. The water coolers now become water-cooled condensers, hence the expansion valves of these units are by-passed. By reversing the flow through the liquid receiver, the process of reversing the refrigerant cycle is completed.

As indicated by the schematic diagram, Fig. 5, this change is easily accomplished. Distinguishing colors have been used for the winter and summer valves, to aid the operator.

TESTS

During the test period in February, 1935, the equipment was operated as a heating plant. The outdoor temperature varied between 33 F and 46 F, hence the equipment was operating between one-third and one-half of its maximum capacity, and the tests cover the case of full capacity operation only for short periods during which the indoor temperature was allowed to rise slightly above normal. However, by determining from these tests the constants of the various units of the equipment, it is possible to calculate accurately the performance at other load conditions. This is true particularly because the temperatures of the water-heated evaporators are confined to a narrow range, and do not depend directly on the temperature of the outdoor air.

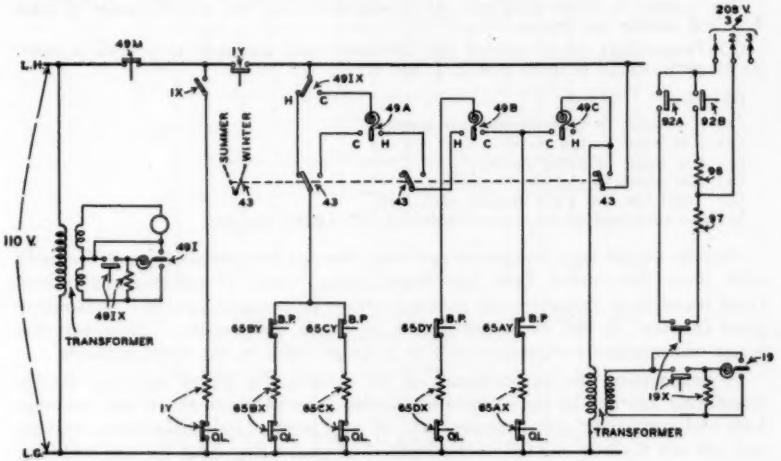
Measurements taken in the tests were:

Water

(a) Rate of flow.

(b) Temperature of water entering and leaving each cooler. These measurements were made by thermo-couples soldered to the water pipes; also by thermometers located in wells, and these readings were checked by bleeding off water at three points (inlet, intermediate and outlet) using both thermo-couple and thermometer measurements.

These readings represent one method of measuring heat absorbed by the evaporators.



INDEX TO SYMBOLS

1X - FAN CONTROL SWITCH	65AX - COMPRESSOR CONTROL CONTACTORS
1Y - " " CONTACTOR	65BX - " " "
19 - HUMIDISTAT	65CX - " " "
19X - HUMIDISTAT RELAY	65DX - " " "
43 - TRANSFER SWITCH	65AY - BACK-PRESSURE CONTACTS ON COMPRESSOR
49A - OUTSIDE THERMOSTAT	65BY - " " " " "
49B - " "	65CY - " " " " "
49C - " "	65DY - " " " " "
49I - INSIDE "	92A - FLOAT SWITCH
49IX - " " RELAY	92B - " "
49M - IMMERSION THERMOSTAT CONTACTS	98 - VALVE SOLENOID
	97 - HEATER COIL

FIG. 11. SCHEMATIC DIAGRAM OF HEATING SYSTEM

Refrigerant

(a) For each of the 4 heat pump units, suction and discharge pressures were read, and converted to equivalent temperatures. Readings of temperature were taken at each evaporator outlet and at each compressor intake and discharge to indicate the degrees of superheat at these points. Temperatures at each condenser inlet, and outlet were recorded, and also liquid temperatures before the expansion valves, to indicate the degree of sub-cooling of the liquid refrigerant. These points determine the refrigerant cycle, and are used in conjunction with standard compressor test data, to determine the heat output of the machines. Incidentally a further check based on compressor speeds, displacements and clearances agreed closely.

Air

(a) Total air flow through the heating coils (refrigerant condensers) was obtained by Pitot tube traverses in the conditioner discharge duct.

(b) The intake of outside air was measured by traverses with an electric hot-wire anemometer, the velocities being too low to permit accurate measurement with a Pitot tube.

(c) Temperature of outside air entering was measured by three thermo-couples and a recorder. An aspirating psychrometer, reading both wet-bulb and dry-bulb temperatures, gave a measure of relative humidity of entering air.

(d) The temperature of air leaving the preheater was recorded, to provide a measure of preheater output.

(e) The condition of recirculated air was measured by means of thermo-couples and an aspirating psychrometer.

The amount of recirculated air can be obtained from the measurements of total flow and outside air intake.

(f) Temperature of air leaving the conditioner was measured to provide a check on the performance of units Nos. 2, 3, and 4.

Electrical

(a) Kw input to each compressor motor.
(b) Kw input to fan motor.
(c) Kw input to pump motor.
(d) Kw input to electric humidifier.
(e) Total kw and kwhr during each test.

Voltage remained at the normal value of 208 during the test.

Outputs of the units as determined from the air measurements agreed closely with those determined from the refrigeration cycle. Capacities determined from water flow measurements checked within 10 per cent, which is reasonably good in view of the small differences in water temperature. However, the water measurements were used only as a rough check on the other methods.

In determining the performance of the system, the power used by the air circulating fan and by the electric humidifier have not been taken into account. Fan and humidifier elements are part of any winter air conditioning system, and are not directly related to the method of generating heat for the building. Using the equipment for heating, rather than cooling, the coefficient of performance has been calculated from the test data on two bases:

(1) (Condenser output plus machine losses) divided by (Electrical input to the compressors).

(2) (Condenser output plus machine losses) divided by (Electrical input to compressors *and* water pump).

The work of compression is included as useful output. The machine losses also may justly be included in the useful heat output since this heat is used to maintain temperature in the basement, which is used only for files and storage, and is not air conditioned.

Method (2) is really the correct criterion of performance, since the pump is an essential part of this particular system. The pump power increases the total power input, but does not increase the heating effect within the building, except by the small amount of the motor losses. Hence, the coefficient of performance is lowered by including this item.

Fig. 12 indicates that, without considering the pump, the coefficient of performance for the composite equipment averages about 3.9, and varies somewhat with outdoor temperature, *i.e.*, with the number of machines in operation. This chart also shows that the performance is best at the extremes of the operating range. In mild weather, only one of the 4 units operates, and the unit having the largest condenser is logically selected for this service, since it can

operate with the lowest refrigerant condensing temperature. This and the high evaporator temperature of a single machine with maximum water flow yield a high coefficient of performance. As units are added for colder weather, more heat is extracted from the water, and some of the units must therefore operate at lower evaporator temperatures with poorer performance at inter-

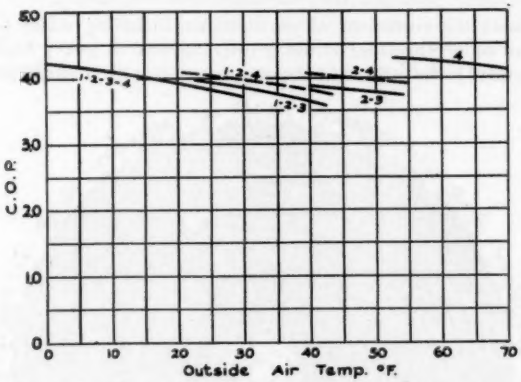


FIG. 12. OVERALL COEFFICIENT OF PERFORMANCE FOR SYSTEM, SALEM HEAT PUMP

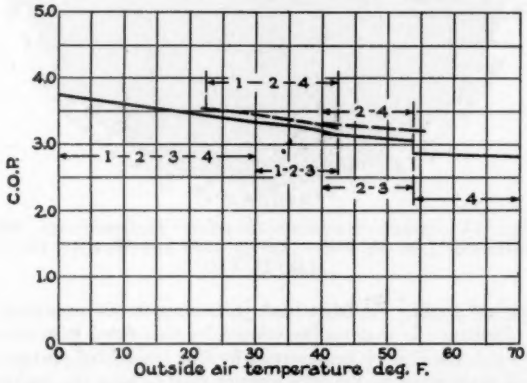


FIG. 13. OVERALL COEFFICIENT OF PERFORMANCE FOR SYSTEM, INCLUDING WATER PUMP INPUT, SALEM HEAT PUMP

mediate outdoor temperatures. However, as lower incoming air temperatures are encountered, the preheater and the units immediately following are able to operate at very low condensing temperatures and the high coefficient of performance of these individual machines raises the average of performance of the group in very cold weather. With air entering at 0 F, the preheater unit alone develops a coefficient of performance of 5.0.

As indicated by the dotted lines of Fig. 12, the order in which machines are added affects the performance by as much as 10 per cent at certain outdoor temperatures. As would be expected, best performance is obtained by using as a base load machine the one having the largest condenser, and by adding machines in such order that the largest remaining condenser surface is associated with the lowest available temperature of entering air.

Fig. 13 shows the coefficient of performance including water pump power. The coefficient of performance is not greatly reduced at heavy loads (low outdoor temperatures) since the water pump input is only a small percentage of

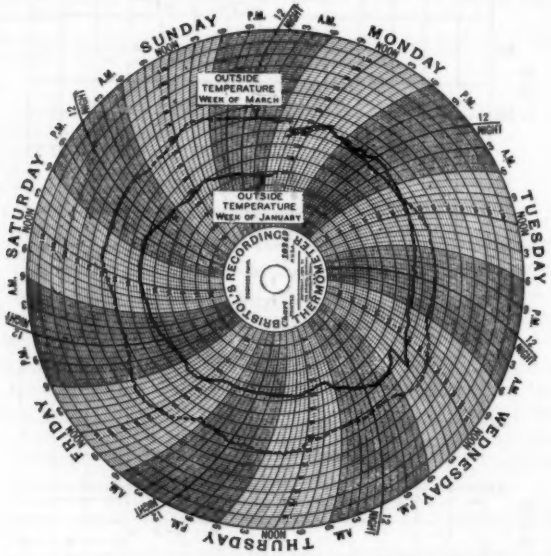


FIG. 14. CHART SHOWING OUTDOOR TEMPERATURES FOR WEEKS OF JAN. 21, 1935—JAN. 28, 1935 AND MAR. 18, 1935—MAR. 25, 1935

the total electrical input. At light load (one compressor operating) the coefficient of performance is materially reduced by this fixed item of water pump power. However, the energy represented by this light load operating condition is only a small percentage of the total energy used to heat the building throughout the winter.

PERFORMANCE

The performance throughout the entire heating season of 1934-35 has been as expected, the equipment functioning perfectly with few minor exceptions enumerated. The lower floor of the building, being the office of an electric utility in a fairly substantial community, presents a rather difficult problem in maintaining constant levels in heating and air conditioning. With people walking in and out almost continuously, and with a wide variation in traffic over a weekly or monthly interval, it would be expected that a particularly heavy

strain would be placed on the regulating and air conditioning equipment. Nevertheless, throughout the entire period an equable and proper temperature was maintained under all conditions of outside temperature. Figs. 14 and 15 illustrate this point very clearly, Fig. 14 representing a week of rather severe weather conditions for the climate during the week January 21-28, 1935, and another week of much milder and more even temperature during the week of March 18-25, 1935. The corresponding indoor temperatures are shown in Fig. 15. As will be noted, in spite of the wide divergence in outside temperatures and some quite low temperatures for the territory, the inside temperatures were

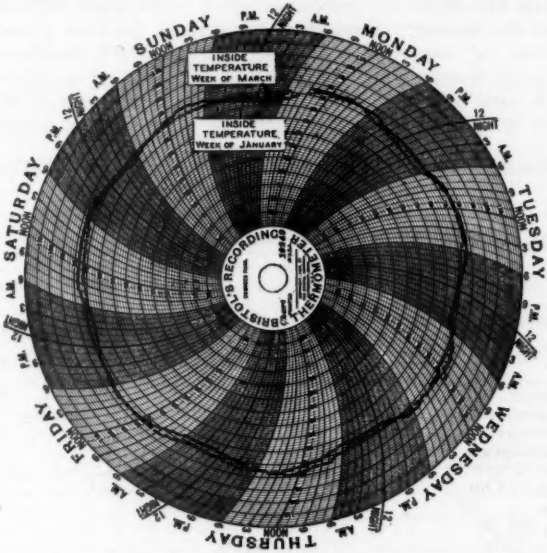


FIG. 15. CHART SHOWING INDOOR TEMPERATURES FOR WEEKS OF JAN. 21, 1935—JAN. 28, 1935 AND MAR. 18, 1935—MAR. 25, 1935

maintained on almost even scales for corresponding periods in the two weeks with no overloading in any way on the equipment.

Although integrated results of the entire winter's operation are not yet available, monthly data up to the end of April have been obtained. The data given in Table 1 covering the month of February, are indicative of a typical month representing neither the extreme low temperature nor the extreme mild weather conditions encountered.

Based upon the data presented as to heat losses in the building, the coefficient of performance is calculated on the basis of both debiting the heat pump with the pump power and on the basis of neglecting this, and gives coefficients of performance of 3.51 and 4.45, respectively. From the data in Fig. 13, it would be expected that the performance under an average condition of 35 deg would be approximately 3.3 as compared with the measured value of 3.51, and the

performance actually obtained is sufficiently close to that, the higher performance being probably due to the effect of solar radiation not taken into account in the calculation of heat losses.

It is expected that the performance at the end of this season will actually be somewhat better than obtained during the last season, due to the fact that the tests have indicated that slight variation in the unit combination employed over the temperature range will give somewhat better performance. This is shown graphically in Fig. 16, which indicates the heating capacity (condenser output of the units) in comparison with the load. The lower load curve represents the heat loss of the air conditioned space. The upper curve includes the heat loss of the basement. With all units operated, the heat given off by the motors, drives and compressors, amounts to about 32,000 Btu/hour, which is very close to the heating requirements of the basement. Fig. 16 also indicates that there is some over-capacity at all times, but the sequence of units reduces this capacity to a relatively small amount. By using a larger number of units,

TABLE 1. ACTUAL PERFORMANCE DATA ON SALEM HEAT PUMP
JANUARY 31 TO FEBRUARY 28, 1935

Outdoor Temperature	Deg
High	55.
Low	10.
Average	34.5
Indoor temperature	
High	78.
Low	56.
Average	70.
Incoming water	Average 57.
Outgoing water	" 53.
Water consumption	915,300 Gal
Electric consumption	Kwhr
Fan motor, 2 hp	582
Pump motor, 3 hp	1502
Humidifier	1503
Compressor, No. 1	2085
" " 2	1370
" " 3	1283
" " 4	1015
Total metered	9475

it is obvious that the capacity of the compressors could be made to conform still more closely to the load, but it was not believed that the additional cost and complication would be justified. However, it will be possible by changing the combinations indicated in solid lines to those indicated in dotted lines, to improve the efficiency and thus bring up the annual coefficient of performance and it is expected to make this change for the next season's heating requirements.

A few minor difficulties were experienced during the winter season. These consisted of ordinary adjustment troubles on compressors, gas joints, belts and pulleys, but all of these were easily corrected. Perhaps the most serious difficulty encountered was the tripping of the water pump motor due to an incorrect operation of the thermal overload relay, resulting in a stoppage of water under a condition where the compressors were running. This resulted in frozen

evaporators and it was necessary to take the compressors out of service for a day to thaw the evaporators out. This, however, has been taken care of by a change in the control wiring, giving the proper interlock to prevent a repetition of this difficulty.

In summer, using only machines Nos. 2 and 3, the system will operate as a conventional direct expansion system with water cooled condensers. These 2 units meet the cooling requirements and will be operated intermittently under room thermostat control with continuous air flow. Neglecting the water pump input, a coefficient of performance of 3.0 is expected; including the water pump, coefficient of performance will be approximately 2.3. These figures are based

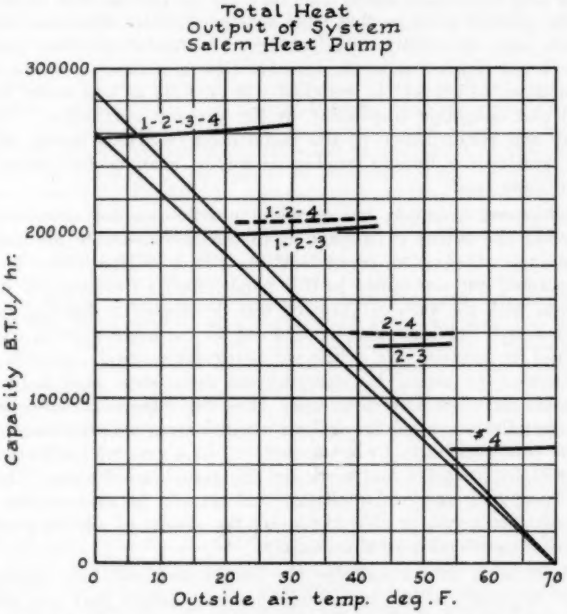


FIG. 16. TOTAL HEAT OUTPUT OF SYSTEM, SALEM HEAT PUMP

on the cooling effect produced, rather than on condenser heat delivery. The kilowatt demand of the system in the summer, including the pump, will be approximately 55 per cent of the winter demand, although the maximum summer load in Btu is only 40 per cent of the maximum heating load.

Due to the many new elements involved which had to be analyzed for this particular installation, it is difficult to evaluate correctly the initial cost of this type of system as compared with more conventional plants. One item, the cost of heat transfer surfaces, is inherently high since the condenser temperature must be low compared to steam coil temperatures to permit efficient operation of the heat pump. However, with the experience gained from this installation, it is expected that the cost of future installations of this type should not greatly

exceed that of equivalent year round systems, which use combustion equipment for heating and mechanical refrigeration for cooling. For installations in milder climates where the maximum cooling and heating loads are more nearly equal, the excess cost of the reverse cycle equipment needed for heating tends to disappear completely.

THE FUTURE

While it was fully realized that the particular working out of the heat pump principle described in this paper would have rather limited direct application, nevertheless it is a fact that there are many cases in this country where such applications can be directly made. In fact, the surprising fact to the authors has been the number of cases that have developed where duplicate applications can be made and, apparently, to advantage. A number of these installations in fact are being figured on at the present time and there is every reason to believe that those which will be installed will be fully as successful, both from a technical and economic standpoint, as the present installation. Experience both in test and performance of the installation described herein dictates no substantial, and only a limited number of minor, changes for future applications of the same type.

The development obviously has excellent possibilities for extensive application. Whether the extent is so large as to render reasonable the looking forward to its application on an economical basis in a million homes is questionable. As pointed out previously in this paper, such an eventuality is unlikely of realization with the present form of this development; but the steps from the form in which this case was carried out to the more general case, where storage would be involved, are, while not particularly simple, nevertheless definitely attainable: the means for carrying out these steps have been definitely indicated from the tests here described. It is the intention, therefore, just as soon as an installation suited for full air conditioning, and representing within itself all the problems likely to be encountered in a general application of that kind, is available, to tackle and work out the complete solution. The authors hope and expect that such an installation will actually be made within the next year and hope further to be able to present the results of such a general solution before the membership of this Society.

The authors wish to acknowledge the cooperation and help received from members of the staffs of the two companies with which they are affiliated in carrying out this installation and in making the tests described and, more particularly, to record their thanks to Mr. A. M. Newman of the Atlantic City Electric Co. for his help in carrying out the tests and in following up the operation of the equipment, and Mr. H. Gibson of the General Electric Co. for his help in working up the test data.

DISCUSSION

H. E. TURNER[7] (WRITTEN): Mr. McLenegan has told you that I would say something about comparative costs of heating by the reverse refrigeration process and by some of the other common methods. It also falls to my share to answer the one question as to how much 100,000 Btu of heat would cost with the heat pump.

[7] American Gas and Electric Co., New York, N. Y.

It is difficult to draw comparisons between the several methods of heating, because first of all the heat pump equipment is designed to take care of heating in the winter, cooling in the summer, and air conditioning the year 'round. Also, up to now at least, the investment in heat pump equipment has been much higher than it will be later on when this system comes into more general use. On the basis of heating alone, however, comparing the costs with those for a modern oil burning installation, a good idea can be given of the relative economy if a few assumptions are made.

Taking a modern 6-room house moderately well insulated as an example, a modern oil burning furnace and the necessary piping and radiation system will cost installed about $1500. The annual carrying charges on this investment, taken at a conservative figure, would be at least 12 per cent, or $180.00 a year. The fuel cost, in the northern states at least, will be about another $150.00 per year, or a total yearly cost for heating of about $330.00. A heat pump to heat such a building would require approximately 150,000,000 Btu for the heating season, which, at a rate of 1 cent per kwhr, and a performance coefficient of about 3, would cost for energy about $150.00, or about the same as the oil burning installation. The investment cost in such an equipment today would, however, be somewhat higher than for an oil burning installation and a full heat pump installation to heat and cool might be approximately $4000. Of this investment possibly $2500 could be considered as chargeable to the heating, with the other $1500 chargeable to summer cooling. The heating investment in this case of $2500 would be considerably more than for modern oil burning equipment, but the cooling investment would not be any more and probably considerably less than the additional investment required in addition to the oil furnace installation to give the same year 'round performance. That is, even today, to accomplish winter heating, summer cooling and all year air conditioning, the investment and the operating cost in the two cases would be pretty close. As the heat pump comes into more general use the equipment associated with this scheme will, of course, become much less expensive and this all-purpose system will have economic and practical advantages over the more commonly accepted and conventional systems. Under such conditions there is the further advantage in favor of the heat pump that the investment is usable all year 'round, whereas in the case of the other combination systems the heating investment and the cooling investment will be idle for almost half of each year.

From the electrical utility viewpoint, installations of the kind described in the paper are very interesting. If electrical heat is ever to be attractive, taking into consideration that the generation of electricity is to a large extent by coal, with an overall efficiency of approximately 35 per cent, the heat pump is a method which seems attractive, trebling or quadrupling the effect of a kilowatthour in terms of heat over what can be done by the consumption of the same kilowatthour in a resistance heater. That is, the heat pump will make a kilowatthour which was generated from coal at about 35 per cent efficiency available as heat in the place where it is to be used as though the original coal had been burned at 100 per cent efficiency, with the practical effect that while carrying this energy over the transmission system it has been reduced in amount about 66 per cent. From the utility standpoint, therefore, this kind of load has the advantage that distribution system capacity only has to be provided for roughly one-third of the kilowatthours that would be provided in a heating system, directly converting kilowatthours to Btu with resistance heaters.

The problem of finding effective sources of heat, regardless of climatic limitations, so that the heat pump principle can be economically and generally applied, will be overcome in time. This method of heating, cooling and air conditioning should eventually be widely accepted, eliminating as it does the nuisances attending the burning of solid and gaseous fuels and also those which would be encountered with the necessary separate air conditioning systems.

In presenting this comparison no consideration was given to heating by solid fuels or even to direct space heating electrically. The solid fuel comparison obviously does not come into the picture. There is not the same justification, however, in leaving out direct space heating, because when the low investment in heating equipment is taken into consideration, there are cases where, with well insulated buildings and surplus seasonal dump or other low cost energy available, this form of heating can make a good economic showing, although this is definitely confined to such rather limited conditions.

Questions have been asked as to rates available for heat pump users. While no regular tariffs, so far as I know, have as yet been established, if the demand becomes great enough, suitable rates will no doubt be made available. At 1 cent per kwhr, this form of heating is decidedly attractive and there are already water heating tariffs in the United States giving rates as low as 1½ cents and 1 cent per kwhr.

As to the cost of 100,000 Btu of heat, using the heat pump principle, if a rate of 1 cent per kwhr is taken, and a coefficient of performance of 3 is assumed, which our operating experience with the system described shows to be conservative, the cost would be approximately 10 cents.

No. 1021

A RATIONAL HEAT GAIN METHOD FOR THE DETERMINATION OF AIR CONDITIONING COOLING LOADS

By F. H. FAUST *(*MEMBER*), L. LEVINE * AND F. O. URBAN * (*NON-MEMBERS*)
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INTRODUCTION

SCIENTIFIC application of air conditioning equipment requires that such equipment shall have adequate capacity to maintain specified indoor conditions, but that the margin of capacity over what is required shall not be so great as to make the installation economically unjustifiable. Thus, the first step in an application is to determine the maximum cooling effect which is necessary, and the second step is to select equipment which is capable of producing this required cooling effect both economically and with a degree of flexibility necessary to meet varying conditions of operation.

This paper summarizes the factors which affect the cooling load, and describes a systematic and rational method for accurately determining its character and magnitude. A heat gain calculation sheet is presented which suitably organizes and tabulates the work in a minimum of space.

The heat gain method to be described presents certain features of advantage to the user, namely:

1. It is applicable to any kind of an enclosure, ranging from a single room to a complete building.
2. It saves time required for making the calculations.
3. It minimizes the chances of error by systematizing the work and providing a suitable calculation form.
4. It permits the engineer to select equipment of adequate capacity, without unnecessary margins, by tabulating accurately each component of the cooling load.
5. It permits the engineer to select the type of control necessary for providing flexibility to meet all conditions of operation by indicating the load components and the manner in which they vary.

The manner in which this method provides the benefits summarized will become apparent in the course of the description.

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June, 1935, by F. O. Urban.

COMPONENTS OF HEAT GAIN

The cooling load is composed of five different components:

1. Heat conducted through walls, windows, etc.
2. Heat absorbed from radiations of the sun.
3. Heat generated by lights and appliances, and other miscellaneous sources.
4. Heat brought in by outdoor air.
5. Heat liberated by people.

The components of heat gain, classified by source, may be further classified as sensible and latent heat gain. The first two components fall into the classification of sensible heat gain; that is, they tend to raise the temperature of the air within the structure. The last three components also produce sensible heat gain, but in addition they may produce latent heat gain; that is, they may tend to increase the moisture content of the air within the structure. Each component is discussed briefly below.

Heat is conducted through walls and partitions because the temperature of the air within the air conditioned space is lower than that on the opposite side of the wall. This is the same process as that by which heat is lost through walls in winter.

Heat absorbed from radiations of the sun increases the cooling load in two ways. In the first place, the sun transmits invisible, but intense, heat rays which increase the temperature of all surfaces exposed to them. With this higher temperature, more heat is conducted to the interior of the structure than if the walls were not exposed to the sunshine. In the second place, the heat rays of the sun pass almost undiminished through ordinary window glass which is exposed to them.

Temperatures of flat roofs have been observed to rise as high as 180 F on a bright summer day, 75 to 80 deg above the temperature of the air. Heat conduction into top floor rooms may be more than doubled because of this effect. Glass windows exposed to the sun may allow as much as 90 per cent of the incident heat to pass through, and it is not uncommon for this portion of the cooling load to be more than the total conduction through the wall proper.

Heat is generated by energy consuming appliances within the air conditioned space, such as electric lights, motors, coffee urns, steam tables, etc. Some of the appliances, such as electric lights and motors, produce only sensible heat gain. Other appliances, such as coffee urns, produce both sensible and latent heat gains. Latent heat gains may arise from the evaporation of moisture within the appliance, or from the liberation of moisture as a product of combustion where a gas flame is present.

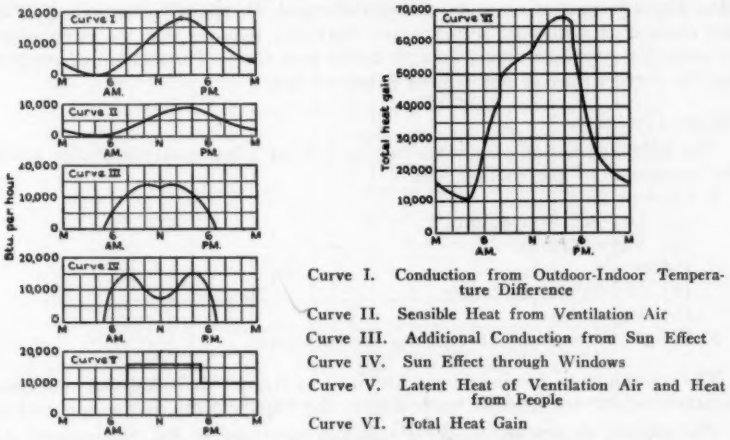
Heat gain results from the incoming outdoor air, which may be introduced by natural, uncontrolled infiltration or by controlled ventilation. This is both sensible and latent in character, as outdoor air entering the structure may have to be both cooled and dehumidified.

Heat is liberated by people through convection and radiation from the surface of the body, and the evaporation of moisture from the skin. The former adds to the sensible cooling load, and the latter to the latent cooling load. Where a number of people are in a relatively small space, the heat they contribute becomes an important item. The rate at which heat is contributed by a normal person not engaged in undue activity may be compared to the absorption of an

average domestic refrigerator, or to that required to completely boil away a pint of water in 3 hours.

Relation Between Total Heat Gain and Individual Components

A complete heat gain calculation to determine the maximum requirements for cooling involves not only the independent calculation of each of the various components, but it involves also the proper combination of these components to determine the correct maximum heat gain. Each individual component varies in magnitude from hour to hour during the day and reaches a maximum or peak value at some particular time. However, the individual maximums do not necessarily occur simultaneously, so that the maximum total heat gain actually may be less than the sum of the maximum values of the several components.



Curve I. Conduction from Outdoor-Indoor Temperature Difference

Curve II. Sensible Heat from Ventilation Air

Curve III. Additional Conduction from Sun Effect

Curve IV. Sun Effect through Windows

Curve V. Latent Heat of Ventilation Air and Heat from People

Curve VI. Total Heat Gain

FIG. 1. TIME VARIATIONS OF COMPONENTS AND TOTAL OF COOLING LOAD IN A SUITE OF OFFICES (EFFECT OF TIME LAG NEGLECTED)

For example, refer to Fig. 1, which shows a family of curves worked up in the calculation of heat gain for a group of offices.

Curves 1 and 2 show the heat conducted through walls, and the sensible heat of ventilation air, respectively. The variations in these components result from changes in the temperature of the outdoor air.

Curve 3 shows the variation in additional conduction through walls exposed to the sun. The intensity of the solar heat striking a wall depends on the angle between the wall and the rays of the sun. This intensity is greatest when the rays of the sun are most nearly perpendicular to the wall, and hence the progression of the sun from east to west causes a variation in this component of the heat gain. Similarly, curve 4 shows the variation in heat gain from solar heat passing directly through windows.

Curve 5 shows the load imposed by the latent heat gain of ventilation air, with the heat from occupants superposed during the period of day when such occupants are present. The reason why the latent heat of ventilation air shows no variation will be discussed later.

Curve 6 shows the total heat gain, which is obtained by adding each of the components in the proper phase relation. An analysis of Fig. 1 shows that whereas individual components attain maximum values at 8 a.m., 10 a.m., 2 p.m., 3 p.m., and 4 p.m., respectively, the total heat gain is a maximum at 4 p.m. Clearly it would not be accurate (though it would be conservative) to find the greatest value that each component attains during the day, and then to add all the maximum values together to find the maximum total gain. The difference between the results obtained by adding the maximum values of the individual components, and combining these components at the time of day when the total is maximum, may be as much as 20 per cent, or as little as 5 per cent. The method of calculation described here is sufficiently flexible to permit taking account of the fact that when one component is at its peak, others may have either passed or not yet reached their peak values. The introduction of this element of additional accuracy and flexibility requires that the calculations be made for a predetermined time of day in each case. The method of predicting the correct time of day will be described later.

Design Conditions

The following design conditions used in making a heat gain calculation affect the magnitude of the cooling load:

1. Outdoor air
   (a) Dry-bulb temperature
   (b) Vapor density
2. Indoor air
   (a) Dry-bulb temperature
   (b) Vapor density
3. Weight of outdoor air entering the enclosure.

The conditions of the indoor air are fixed by the dry-bulb and wet-bulb temperatures which are specified according to the requirements of the application.

The amount of ventilation air is specified according to the requirements of the application if ventilation is controlled. Otherwise it is determined by the infiltration rate.

Design outdoor conditions are based on the Weather Bureau records of the locality under consideration. The design outdoor dry-bulb temperature is usually specified as a maximum value. This design temperature for comfort applications may be selected so that it will be exceeded by not more than about 10 per cent of the daily maximum outdoor temperatures during the cooling seasons over a period of several years. Systems designed on the basis of such an outdoor temperature will, on occasional extreme days, be incapable of maintaining normal indoor conditions, but will be generally satisfactory, considering both performance and investment. There are several references [1, 2] on the subject of design outdoor conditions, and recommended values are given in Chapter 8 of THE A. S. H. V. E. Guide.[3]

The design outdoor temperature described is really the design maximum outdoor temperature. As has already been described, the maximum total heat gain

[1] The Relation of Climate to Air Conditioning Design, by O. W. Armspach, *Heating and Ventilating*, Vol. 29, Dec. 1932, p. 26.
[2] Climate in Air Conditioning, by H. W. Skinner, *Heating and Ventilating*, Vol. 29, Dec. 1932, p. 23.
[3] AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE, 1935.

may occur at some time of day when the outdoor temperature is considerably lower than its maximum value. A night club is an outstanding example of an application in which this condition arises. Hence, it is necessary to know not only the design maximum outdoor temperature, but also how the outdoor temperature varies from hour to hour, and how far it may have dropped by evening following a *maximum* day. A careful study of Weather Bureau data indicates that for practical purposes the outdoor temperature may be assumed to vary in the manner shown by Fig. 2. It has been found that on the average the outdoor temperature is highest about 3 p.m., and lowest about 5 a.m. The difference between the maximum and minimum outdoor temperatures on a *maximum* day may be taken as the mean difference between the daily maximum and daily minimum outdoor temperatures for the month of July. This mean difference is termed by the Weather Bureau the *mean daily range* for July. It

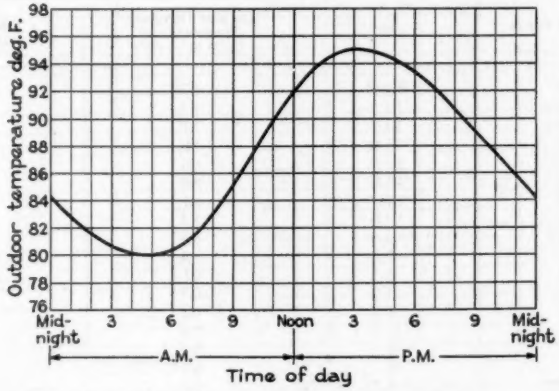


FIG. 2. TYPICAL VARIATION OF OUTDOOR TEMPERATURE

varies from about 10 F in some parts of the United States to about 40 F in other localities. A table of outdoor temperatures for various hours of the day may then be constructed by using the formula:

$$t_o = t_d - B\,V \tag{1}$$

where

t_o = design outdoor temperature at a particular time of day.
t_d = design maximum outdoor temperature.
V = mean daily range for July.
B = an hourly factor, determined by the shape of the curve in Fig. 2, and tabulated in Table 1.

Table 1 shows design outdoor temperatures, calculated as outlined above, for various hours of the day in a locality where the design maximum outdoor temperature is 95 F and the mean daily range for July is 15 F. Of course there are occasional days when the temperature range is less than the mean value, and the temperature is somewhat higher in the evening than is indicated by the above analysis. This possibility should be kept in mind.

It was implied previously that the latent heat gain of ventilation air is substantially constant for various hours of the day. A study of Weather Bureau Data shows that in general the outdoor relative humidity is lowest when the outdoor dry-bulb temperature is highest, and vice versa. Furthermore, this variation in outdoor relative humidity is of such a nature that the vapor density, or moisture content, of outdoor air is approximately constant. Consequently, the outdoor vapor density may be determined from the psychrometric chart once the design maximum outdoor dry-bulb and wet-bulb temperatures are known. The design outdoor wet-bulb temperature may be selected on the same

TABLE 1. DESIGN OUTDOOR TEMPERATURES FOR VARIOUS HOURS OF THE DAY

(t_d = 95 F; V = 15 F)

HOURLY FACTOR B	BV	DESIGN OUTDOOR TEMPERATURE ($t_o = t_d - BV$)	TIME OF DAY
0.8	12.0	83	1 a.m.
0.9	13.5	82	2
0.95	14.2	81	3
1.0	15.0	80	4
1.0	15.0	80	5
1.0	15.0	80	6
0.9	14.2	81	7
0.8	12.0	83	8
0.65	9.8	85	9
0.5	7.5	88	10
0.3	4.5	91	11
0.2	3.0	92	Noon
0.1	1.5	94	1 p.m.
0	0	95	2
0	0	95	3
0	0	95	4
0.05	0.75	94	5
0.1	1.5	94	6
0.2	3.0	92	7
0.3	4.5	91	8
0.4	6.0	89	9
0.5	7.5	88	10
0.6	9.0	86	11
0.7	10.5	85	Midnight

basis as the design outdoor dry-bulb temperature; that is, high enough so that it will include about 90 per cent of the daily maximum wet-bulb temperatures during the cooling season.

The outdoor vapor density is not exactly constant during the entire day, and of course shows sharp variations during stormy periods. The assumption of a constant design outdoor vapor density on a *maximum* day, however, is reasonably accurate for comfort applications. It must be recognized that all assumptions in regard to climatic conditions are of necessity approximate.

Time Lag Resulting from Heat Storage

The heat storage capacity of walls results in a phenomenon akin to a time lag in the flow of heat through them. The result is that the maximum effects of

Name _____ Room used for _____ Job No. _____

Street _____ Room No. _____ Floor _____ Sheet No. _____

City _____ Calculated by _____ Date _____

DESIGN CONDITIONS

Outdoor: D.B. Temp. _____ °F. W.B. Temp. _____ °F. Abs. Hum. _____ Gr. per lb. Calculations are for _____ a.m. p.m.

Indoor: D.B. Temp. _____ °F. W.B. Temp. _____ °F. Abs. Hum _____ Gr. per lb. Volume of room _____ cu. ft.

Temp. Diff. _____ °F. Hum. Diff. _____ Gr. per lb. Infiltration rate _____ cfm.

Ventilation Air, Required Amount _____ cfm. (at indoor design conditions).

SENSIBLE HEAT GAIN

Column	I	II	III	IV	V	VI	VII			VIII	IX	
SIDE	Direction Wall Faces	Wall or Partition Dimensions / Windows and Doors, Number and Dimensions	Gross Wall Area, Sq. Ft. / Total Glass Area, Sq. Ft. / Net Wall Area, Sq. Ft.	— / U_g / U_w	Design D.B. Temp. Diff. °F.	Conduction (col. Temp. Diff.) Btu. per hr. III × IV × V	Calculation of R_w for VIII $R_w = F \times A \times I$			— / R_g / R_w	Sun Effect Btu. per hr. III × VIII	REMARKS
							F	A	I			
1st				—		—						
2nd				—		—						
3rd				—		—						
4th				—		—						
5th				—		—						
Ceiling				—		—						
Floor				—		—						

Total Conduction _____ Total Sun Effect _____

Conduction: Sum of Col. VI ..... = _____ Btu. per hr.

Ventilation air: Cfm. (_____) × Temp. dif. (_____) × 1.08 ..... = _____

Sun effect: Sum of Col. IX ..... = _____

People: No. of persons in room (_____) × 230 ..... = _____

Electrical appliances: Watts (_____) × 3.4 ..... = _____

Other sources: Btu. per hr. ..... = _____

TOTAL SENSIBLE HEAT GAIN (Without window shading) ..... = _____ Btu. per hr.

Saving from awnings or shades: Total glass sun effect (_____) × shading factor (_____) = _____

TOTAL SENSIBLE HEAT GAIN (With window shading) ..... = _____ Btu. per hr.

LATENT HEAT GAIN

Ventilation air: Cfm. (_____) × Hum. dif. (_____) × 0.68 ..... = _____ Btu. per hr.

People: No. of persons in room (_____) × Btu. per hr. per person (_____) ..... = _____

Other sources: Lb. of water per hr. (_____) × 1060 ..... = _____

TOTAL LATENT HEAT GAIN ..... _____ Btu. per hr.

TOTAL SENSIBLE HEAT GAIN ..... _____

TOTAL HEAT GAIN _____ Btu. per hr.

EXPLANATIONS

U—Transmission Coefficient.
R—Solar Radiation Coefficient.
I—Intensity of Sun.
A—Absorption Coefficient.
F—Radiation Factor.
w (subscript)—Wall.
g (subscript)—Glass.

See Engineering Data Book for coefficients and factors.

Heat from People, Btu. per hr. per person:
Average condition (at rest, eating, office work, etc.)
Sensible 230
Latent 190
Medium rate of exertion (restaurant waiters, average dancing, etc.)
Sensible 230
Latent 420

FIG. 3. HEAT GAIN CALCULATION SHEET

outdoor temperature and solar radiation may not manifest themselves on the interior until some time after the actual maximum outdoor conditions have passed. This introduces the problem of determining time lag, which will be discussed later.

Summary of Analysis

The preceding paragraphs have discussed briefly the five components of the

heat gain which were summarized in the beginning, and further classified them as sensible and latent. These components have been shown to vary during the day in a manner dependent on climatic and indoor conditions. This causes a phase relation between them of such a nature that the correct maximum heat gain is not necessarily the sum of the maximum values of the components, but is the sum taken at the particular time of day when the total is a peak. This is further affected by time lag in walls.

Design conditions which affect the character and magnitude of the heat gain have been listed, and a method suggested for determining design outdoor conditions. The determination of design indoor conditions has been omitted because they are usually specified by the requirements of the application, and depend on a number of factors which are often beyond the control of one who is making the heat gain calculation.

The following sections of this paper will be devoted to methods of determining accurately the character and magnitude of the components of the heat gain, and of combining them properly to predict the total cooling load.

CALCULATION OF THE COMPONENTS OF HEAT GAIN

Heat Gain Calculation Sheet

A calculation sheet for organizing and tabulating the calculations described in the following paragraphs is shown in Fig. 3.

Conduction from Temperature Differences

The technique of calculating heat conduction through a wall is well known and widely used in determining heating requirements for winter. The heat gain by conduction is calculated from the formula:

$$H_c = A_w\, U_w\, (t_o - t_i) \qquad (2)$$

where

H_c = heat conducted through the wall, window or other partition, Btu per hour.
A_w = Area of wall, window, etc., sq ft.
t_i = design indoor temperature, F.
t_o = design outdoor temperature, F.
U_w = overall coefficient of heat transmission of the wall, window, etc., Btu per hour per sq ft per F.

A complete discussion of the transmission of heat through walls, etc., is given in THE A. S. H. V. E. GUIDE,[4] Chapter 5, together with transmission coefficients for walls of various types of building construction. Special cases, which arise in connection with rooms under attics and sloping roofs, are covered in Table 2 of this paper.

In making this calculation, care must be exercised to consider separately each section of the wall having different overall heat transmission coefficients or different temperature differences. For example, the overall coefficient for a window is different from that for the wall in which it is located. This is taken care of automatically on the calculation sheet.

Careful judgment must be exercised in selecting temperature differences. Thus, the value for outside walls is equal to the difference between the design outdoor temperature, and the design indoor temperature, while the temperature

[4] Loc. Cit. See Note 3.

difference across inside partitions may be greater or less. For inside partitions, with no unusual sources of heat on the far side, it is justifiable to assume temperature differences from 3 to 5 deg less than for outside walls, at that time of day when the outdoor temperature is highest. If the partition adjoins a hot kitchen, or similar heated space, it is desirable to assume a temperature difference somewhat greater than for the outside walls.

Sun Effect Through Walls

The heat absorbed from solar radiation falling on exposed surfaces has already been discussed briefly. The actual process that occurs can be explained

TABLE 2. TRANSMISSION COEFFICIENTS, TEMPERATURE DIFFERENCES AND SOLAR RADIATION COEFFICIENTS FOR CEILINGS

LOCATION OF ROOM	U_w	TEMP. DIFF.	R_w
Not on top floor	$U_w = U_c$	$(t_o - t_i) - (3 \text{ to } 5 \text{ Deg})$	$R_w = 0$
Under unventilated attic with sloping roofs (ordinary house construction)	$U_w = \dfrac{U_c}{1 + 0.7\dfrac{U_c}{U_r}}$	$(t_o - t_i)$	$R_w = FaI$
Under unventilated attic with flat roof	$U_w = \dfrac{U_c}{1 + \dfrac{U_c}{U_r}}$	$(t_o - t_i)$	$R_w = FaI$
Under well ventilated attic (natural ventilation)	$U_w = U_c$	(Attic Temp. $- t_i$)	$R_w = 0$
Under fan ventilated attic	$U_w = U_c$	$(t_o - t_i) + (5 \text{ to } 10 \text{ Deg})$	$R_w \neq 0$
Directly under flat roof (roof also forms the ceiling)	$U_w = U_r$	$(t_o - t_i)$	$R_w = FaI$
Directly under sloping roof (roof also forms the ceiling)	$U_w = \dfrac{\text{Roof area}}{\text{Horizontal Ceiling Area}} U_r$	$(t_o - t_i)$	$R_w = FaI$

Notes:

1. t_o = Design outdoor dry-bulb temperature.
2. t_i = Design indoor dry-bulb temperature.
3. U_w = Overall transmission coefficient.
4. U_c = Transmission coefficient for ceiling alone.
5. U_r = Transmission coefficient for roof alone.
6. F = Radiation factor. Determine F to use in fourth column from Fig. 5 by using the value of U_w calculated in the second column.
7. a = Absorption coefficient. (See Table 4.)
8. I = Intensity of sun. Determine I to use in fourth column from Table 3, using values for horizontal surfaces.

simply in connection with Fig. 4. The amount of heat received by each square foot of the wall surface depends upon the intensity of the solar heat and the angle between the surface and the rays of the sun. However, a part of this heat is reflected directly back into space without affecting the temperature of the wall surface, because all surfaces act as heat mirrors to a greater or lesser extent. The nature of the outside surface of the wall determines its effectiveness as a heat mirror, and thus is an important factor in determining how much of the incident solar heat penetrates the wall and adds to the cooling load. Of the heat absorbed by the outside surface of the wall, a portion is dissipated to the outside air and surroundings by convection and radiation and the balance is conducted through the wall into the air conditioned space.

The actual *total* amount of heat conducted through a wall exposed to the sun depends not only on the amount of solar heat which is absorbed by the outside surface but also on the overall heat transmission coefficient, and the temperatures of the indoor and outdoor air. A mathematical expression for the heat balance involved shows that the total heat conducted to the interior may, for practical purposes, be expressed by,

$$H_t = A_w U_w (t_o - t_i) + A_w R_w \tag{3}$$

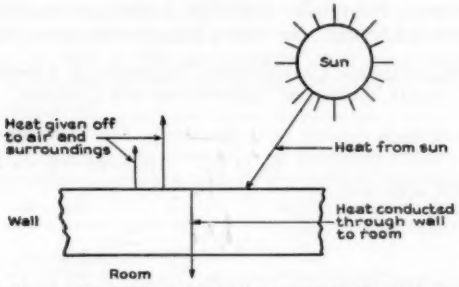


FIG. 4. PROCESS OF HEAT ABSORPTION FROM SOLAR RADIATION

The first term of (3) is identical with (2). The second term is

$$H_r = A_w R_w \tag{4}$$

where

H_r = Additional heat conducted through a wall exposed to the sun, Btu per hour

$R_w = FaI$. (5)

I = Actual solar radiation striking the wall, Btu per hour per sq ft.

a = Percentage (expressed as a decimal) of the incident solar radiation which is absorbed by the wall surface.

F = Percentage (expressed as a decimal) of the absorbed solar radiation which is transmitted to the inside.

It will be noted in (3) that the total conduction through a wall includes two additive terms. The first one is identically the same as that given above for heat conducted through a wall not exposed to the sun. The second term depends only on the intensity of the sun and the characteristics and orientation of the wall, and is independent of the temperature difference across the wall. Thus, equation (4) gives the *additional conduction* through walls exposed to the sun.

The radiation factor R_w depends on I, the amount of solar heat incident on the wall; a, the absorption coefficient of the wall; and F, a factor which gives the percentage of the absorbed solar heat which is transmitted to the interior.

The intensity of solar radiation, I, on a given wall, depends on the amount of water vapor and dust in the atmosphere [5] through which the solar heat must travel before it reaches the surface, and the angle between the rays of the sun and the surface.[6, 7] Table 3 gives values of I for surfaces facing different

[5] The Determination of Sun Effect on Summer Cooling Loads, by Hendrickson and Walker, *Heating and Ventilating*, Vol. 30, June 1933, p. 20.

[6] Summer Cooling for Comfort as Affected by Solar Radiation, by Hendrickson and Walker, *Heating and Ventilating*, Vol. 29, Nov. 1932, p. 14.

[7] Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet and J. L. Blackshaw, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 101.

directions, at different hours of the day and for several northern latitudes. These values of I take into account the average amount of water vapor and dust in the atmosphere throughout the United States, and are applicable for the period of the year during which the heat gain is normally a maximum, namely, from early May to the middle of August.

Average values of the absorption coefficient, a, are given in Table 4. The factor F depends on the overall transmission coefficient, U_w of the wall. Values of this factor are determined from the curve of Fig. 5.

It should hardly be necessary to point out here that the calculations of sun effect through walls described above are made only when the walls are definitely exposed to the sun. If the wall is shaded by trees or other buildings, there will be no heat from the sun striking it directly, and the calculation of this component of the heat gain is omitted. When the wall is only partly shaded, it will be necessary to estimate, or calculate from the geometry involved [8] what percentage of the wall is thus protected, and to make allowances accordingly.

Certain special cases arise in connection with rooms located under attics, or under flat or sloping roofs. Table 2 gives instructions for determining R_w to use in such cases. These are based on using the horizontal ceiling area to determine total sun effect through the ceiling.

The method of calculating sun effect through walls described in this paper makes possible an accurate and scientific determination of this component. Fundamentally, this method is equivalent to calculating the temperature differences across walls exposed to the sun, and avoids the necessity for estimating them. The mathematical elimination of the wall surface temperature, however, results in the simple expressions given by equations (3), (4), and (5). Furthermore, when the above method is used in conjunction with the proper combination of the several components, sun effect is included for only those walls exposed to the sun when the total cooling load is a maximum.

Sun Effect Through Windows

Transparent windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass through undiminished. The actual amount of incident energy which is transmitted depends on the actual transparency of the window glass to the solar heat rays and on the angle between the rays and the surface of the glass.[9, 10] That is, the amount of heat reflected by the exterior surface of the glass depends upon the angle between the glass and the rays of the sun. An ordinary window glass which is exactly perpendicular to the rays of the sun actually permits about 90 per cent of the energy to pass through unobstructed.[11, 12] The heat gain through windows exposed to the sun is calculated by the formula:

$$H'_r = A_g R_g \qquad (6)$$

where

H'_r = solar radiation transmitted through a window, Btu per hour.
A_g = area of the glass, sq ft.
R_g = amount of solar heat transmitted directly, through the glass, Btu per hour per sq ft.

[8] Loc. Cit. See Note 4.
[9] Loc. Cit. See Note 6.
[10] Loc. Cit. See Note 7.
[11] Surface Absorption of Heat from Solar Radiation, by Hechler and Queer, *Refrigerating Engineering*, Vol. 25, Feb. 1933, p. 86.
[12] Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, p. 93.

TABLE 3. SUN EFFECT COEFFICIENTS AT VARIOUS LATITUDES FOR WALLS FACING SEVERAL DIRECTIONS

I = Btu per hour striking 1 sq ft of wall surface
R_g = Btu per hour transmitted by 1 sq ft of single glass (for double glass multiply values of R_g given below by 0.9)

North Latitude, Degrees	Coefficient	Time of Day	Direction Wall Faces							Horizontal Surface
			NE	E	SE	S	SW	W	NW	
20	I, for walls and roofs	6 A.M.	100	105	50					15
		7	165	185	95					70
		8	175	205	115					140
		9	150	185	110					205
		10	105	135	90					260
		11	55	70	50					290
		Noon								300
		1 P.M.					50	70	55	290
		2					90	135	105	260
		3					110	185	150	205
		4					115	205	175	140
		5					95	185	165	70
		6					50	105	100	15
	R_g, for windows and skylights	6 A.M.	95	100	40					
		7	155	170	80					50
		8	155	190	95					120
		9	125	165	85					185
		10	75	110	60					235
		11	25	40	20					265
		Noon								275
		1 P.M.					20	40	25	265
		2					60	110	75	235
		3					85	165	125	185
		4					95	190	155	120
		5					80	170	155	50
		6					40	100	95	
25	I, for walls and roofs	6 A.M.	115	120	60					20
		7	165	190	105					75
		8	165	210	125					145
		9	135	185	125					210
		10	90	135	105	10				260
		11	35	70	65	25				290
		Noon			20	30	20			300
		1 P.M.				25	65	70	35	290
		2				10	105	135	90	260
		3					125	185	135	210
		4					125	210	165	145
		5					105	190	165	75
		6					60	120	115	20
	R_g, for windows and skylights	6 A.M.	105	115	45					
		7	150	175	85					55
		8	150	190	105					125
		9	110	165	100					190
		10	60	110	75					235
		11	10	40	35					265
		Noon								275
		1 P.M.					35	40	10	265
		2					75	110	60	235
		3					100	165	110	190
		4					105	190	150	125
		5					85	175	150	55
		6					45	115	105	

TABLE 3 (*Continued*)

NORTH LATITUDE, DEGREES	COEFFICIENT	TIME OF DAY	DIRECTION WALL FACES							HORIZONTAL SURFACE
			NE	E	SE	S	SW	W	NW	
30	I, for walls and roofs	6 A.M.	125	135	65					25
		7	165	195	110					80
		8	160	210	135					150
		9	125	185	140	10				210
		10	75	135	120	30				255
		11	20	70	85	45				285
		Noon			35	50	35			295
		1 P.M.				45	85	70	20	285
		2				30	120	135	75	255
		3				10	140	185	125	210
		4					135	210	160	150
		5					110	195	165	80
		6					65	135	125	25
	R_g, for windows and skylights	6 A.M.	115	125	50					10
		7	150	180	95					60
		8	140	190	115					125
		9	95	165	110					190
		10	45	110	90	10				235
		11		40	55	20				260
		Noon			10	20	10			270
		1 P.M.				20	55	40		260
		2				10	90	110	45	235
		3					110	165	95	190
		4					115	190	140	125
		5					95	180	150	60
		6					50	125	115	10
35	I, for walls and roofs	6 A.M.	135	145	70					30
		7	160	200	120					85
		8	150	210	145					150
		9	110	185	150	30				205
		10	60	135	135	55				250
		11		70	100	70				280
		Noon			55	75	55			290
		1 P.M.				70	100	70		280
		2				55	135	135	60	250
		3				30	150	185	110	205
		4					145	210	150	150
		5					120	200	160	85
		6					70	145	135	30
	R_g, for windows and skylights	6 A.M.	120	135	55					15
		7	145	185	100					65
		8	130	190	125					130
		9	85	165	125					185
		10	30	110	105	25				230
		11		40	70	40				255
		Noon			25	45	25			265
		1 P.M.				40	70	40		255
		2				25	105	110	30	230
		3					125	165	85	185
		4					125	190	130	130
		5					100	185	145	65
		6					55	135	120	15

TABLE 3 (*Continued*)

NORTH LATITUDE, DEGREES	COEFFICIENT	TIME OF DAY	DIRECTION WALL FACES							HORIZONTAL SURFACE
			NE	E	SE	S	SW	W	NW	
40	I, for walls and roofs	6 A.M.	140	155	80					35
		7	160	200	125					90
		8	140	210	155	10				150
		9	100	185	160	45				200
		10	40	135	150	75				240
		11		70	115	95	15			270
		Noon			70	100	70			280
		1 P.M.			15	95	115	70		270
		2				75	150	135	40	240
		3				45	160	185	100	200
		4				10	155	210	140	150
		5					125	200	160	90
		6					80	155	140	35
	R_s, for windows and skylights	6 A.M.	130	140	65					20
		7	145	185	105					70
		8	120	190	135					130
		9	70	165	140	20				180
		10	15	110	125	45				220
		11		40	90	65				250
		Noon			40	70	40			255
		1 P.M.				65	90	40		250
		2				45	125	110	15	220
		3				20	140	165	70	180
		4					135	190	120	130
		5					105	185	145	70
		6					65	140	130	20
45	I, for walls and roofs	5 A.M.	70	70	25					
		6	145	165	85					40
		7	155	205	130					95
		8	130	210	165	25				145
		9	85	180	175	65				195
		10	25	135	160	95				230
		11		70	130	115	30			260
		Noon			90	125	90			265
		1 P.M.			30	115	130	70		260
		2				95	160	135	25	230
		3				65	175	180	85	195
		4				25	165	210	130	145
		5					130	205	155	95
		6					85	165	145	40
		7					25	70	70	
	R_s, for windows and skylights	5 A.M.	65	65	20					
		6	135	150	70					25
		7	140	190	115					70
		8	110	190	145					125
		9	55	165	150	35				175
		10		110	140	65				215
		11		40	105	85	10			240
		Noon			55	95	55			245
		1 P.M.			10	85	105	40		240
		2				65	140	110		215
		3				35	150	165	55	175
		4					145	190	110	125
		5					115	190	140	70
		6					70	150	135	25
		7					20	65	65	

Values of the radiation coefficient, R_g, for windows in walls facing different directions, at different hours of the day and for several northern latitudes, are given in Table 3. These values of R_g are based on the values of I given in the same table and the reflective and absorptive characteristics of ordinary window glass at various angles between the solar rays and the glass.

Recently there have been developed several special kinds of glass which are more absorptive to solar heat than ordinary window glass.[13] If the structure is

TABLE 4. SOLAR ABSORPTION COEFFICIENTS (a) FOR DIFFERENT BUILDING MATERIALS

SURFACE MATERIAL	ABSORPTION COEFFICIENT (a)
VERY LIGHT COLORED SURFACES	
Such as	
White stone	
Very light colored cement	0.4
White or light cream-colored paint	
MEDIUM DARK SURFACES	
Such as	
Asbestos shingles	
Unpainted wood	
Brown stone	0.7
Brick and red tile	
Dark-colored cement	
Stucco	
Red, green or gray paint	
VERY DARK COLORED SURFACES	
Such as	
Slate roofing	0.9
Tar roofing materials	
Very dark paints	

glazed with windows of this character, the values of R_g given in Table 3 should be reduced in varying percentages depending on the nature of the glass used.

When windows are completely shaded from the sun by trees or other buildings, then none of the solar energy passes through them. If they are partially shaded the percentage reduction must be estimated, or calculated from the geometry involved. This is discussed by Hendrickson and Walker.[14] In the absence of such accidental shading, there are various man-made devices such as awnings, shades, and venetian blinds which may be used to mitigate the effect of the sun to a greater or lesser extent. It is somewhat difficult to give an exact appraisal of the relative effectiveness of these various devices, because as yet complete experimental evidence is not available. However, the matter may be summarized about as follows:[15]

1. Devices hung *inside* the window, such as shades or blinds, are not as effective as devices hung outside. This is because they interrupt the rays of the sun only after they have passed through the glass. Although some

[13] Windows—and Their Relation to Air Conditioning Problems, by W. W. Shaver, *Refrigerating Engineering*, Vol. 26, Sept. 1933, p. 133.
[14] Loc. Cit. See Note 5.
[15] Loc. Cit. See Note 7.

of the heat may be reflected back through the glass, a portion of it is absorbed and transmitted into the room.

2. Awnings are probably not so effective on the first floor of a building as they are on upper floors. In the former case the surface of the ground no doubt reflects some of the solar heat through the opening under the awning, and there is secondary radiation from the warm awning itself.
3. Awnings on the first floor may be assumed to eliminate from the heat gain approximately 75 per cent of the solar radiation which would otherwise pass through the bare window, and awnings on upper floors may be

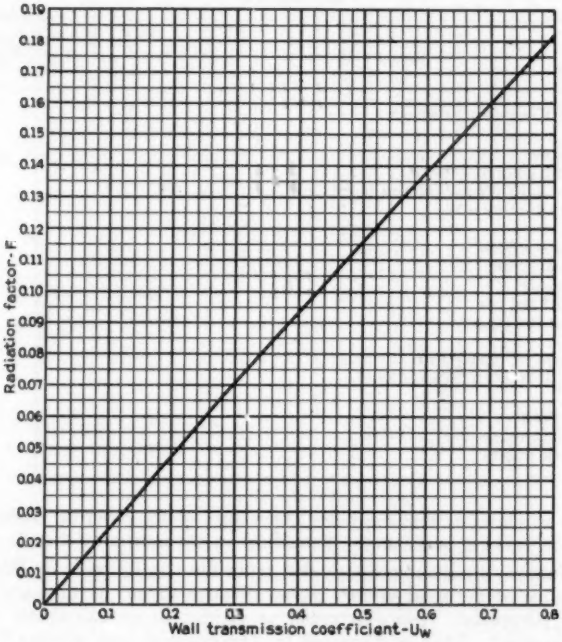


FIG. 5. SOLAR RADIATION FACTOR VS. OVERALL WALL TRANSMISSION COEFFICIENT

assumed to exclude 85 per cent of the solar radiation from the window.

4. Inside shades, buff-colored, clean, and completely drawn, may be expected to eliminate about 50 per cent of the solar radiation which would otherwise pass through the window. Dark shades are not nearly so effective.
5. Inside venetian blinds with metallic aluminum surfaces are about as effective as inside buff shades.

Heat from Lights and Appliances

Heat dissipating appliances which give off only sensible heat may be divided into three classes:

1. Those whose electrical watts input are known, or can be read from a nameplate.
2. Those whose electrical watts input may be determined, through brief calculations.
3. Those which have no electrical input.

In the first category may be found such devices as electric lights, toasters, waffle irons, etc., for which the nameplate indicates the electrical power consumption. All of this power is changed into heat and given off to the air in the room. The amount of heat in Btu per hour thus liberated is equal to 3.4 times the watts input (one watt is equivalent to 3.415 Btu per hour).

Under the second heading may be included such devices as motors which show on their nameplate an hp rating but do not give the watts input for that

TABLE 5. HEAT GENERATED BY MOTORS AND MOTOR GENERATORS OPERATING CONTINUOUSLY

NAMEPLATE RATING IN HP	HEAT GAIN IN BTU PER HOUR PER HP RATING	
	Connected Load in Same Room	Connected Load Outside of Room
1/8 to 1/2	4,250	1,700
1/2 to 3	3,700	1,150
3 to 20	2,950	400

MOTOR GENERATORS

NAMEPLATE RATING IN KILOWATTS	HEAT GAIN IN BTU PER HOUR PER KW RATING
1/4 to 3	2,800
3 to 5	1,300

rating. If the motor efficiency is known, the watts input may be calculated from the formula:

$$P = \frac{746\ (hp)}{n} \tag{7}$$

where

P = motor input, watts.
(hp) = motor load, horsepower.
n = motor efficiency (expressed as a decimal).

If there is no way of determining the motor efficiency, then as a fair approximation, the heat gain from such sources may be obtained by using the data in Table 5, multiplying the figures in the table by the motor hp rating.

The third class of appliances includes items such as ranges, steam tables, coffee urns, uncovered steam pipes, etc., where no electrical power consumption is involved. For such cases it is necessary to make a direct estimate of the heat gain in Btu per hour. Table 6 gives heat from various types of appliances.

There are some appliances which give off not only sensible heat, but latent heat as well. These will be encountered most generally in industrial applications.

Any device which evaporates moisture into the air falls into this category, and the moisture thus evaporated must be included in the calculations. First, calculate the amount of moisture which is evaporated. The latent heat gain from such appliances is then:

$$H_l = 1060\ M \tag{8}$$

where

H_l = latent heat of appliance, Btu per hour.
M = rate of evaporation of water, lb per hour.
1060 = average latent heat of vaporization of water vapor at air conditioning temperatures.

TABLE 6. HEAT FROM APPLIANCES

NATURE OF APPLIANCE	BTU PER HOUR	
	Not Hooded	Hooded and Ventilated
Coffee Urns and Percolators		
1 gallon capacity	1,000	250
2 " "	2,000	500
3 " "	3,000	750
4 " "	4,000	1,000
5 " "	5,000	1,250
10 " "	16,000	4,000
Dish warmers per 10 sq ft of shelf	6,000	
Restaurant range—4 burner and oven	100,000	
Residence electric range		
Small burner	3,400	
Medium burner	4,100	
Large burner	7,700	
Oven	10,200	
Appliance connection	2,200	
Warming compartment	1,000	
Residence gas range		
Giant burner	12,000	
Medium burner	9,000	
Pilot	250	
Oven	1,000 Btu per cu ft of oven volume	

A pitfall to avoid in this calculation is the inclusion of latent heat from appliances in two different parts of the calculation. Thus, suppose an electric hot plate rated at 500 watts were used to boil water, resulting in the evaporation of a half pound an hour. The total heat dissipated by the hot plate would be $500 \times 3.4 = 1700$ Btu per hour. The latent heat given off would be $0.5 \times 1060 = 530$ Btu per hour. The thing to note is that the 530 Btu is included in the 1700 Btu given off by the hot plate. Thus,

Latent heat from the hot plate = 530 Btu per hour.
Sensible heat from hot plate = 1170 Btu per hour.
Total heat from hot plate = 1700 Btu per hour.

Heat from Ventilation Air

Outdoor air may be introduced to an air conditioned space either by natural and uncontrolled infiltration or by controlled ventilation. Such ventilation air must be cooled to room temperature and dehumidified to room humidity. The sensible and latent heat gains from ventilation air are determined accurately from the two formulas:

$$H_s = \frac{60\,Q}{v}\,0.245\,(t_o - t_i) \tag{9}$$

$$H_l = \frac{60\,Q}{v}\,\frac{\omega_o - \omega_i}{7000}\,1060 \tag{10}$$

where

H_s = sensible heat, Btu per hour.
H_l = latent heat, Btu per hour.
Q = ventilation air rate, cfm.
v = specific volume of ventilation air, cu ft per lb of dry air.
ω_o = vapor density of outdoor air, grains of moisture per lb of dry air.
ω_i = vapor density of indoor air.

If Q is referred to average design indoor conditions (80 F, 50 per cent relative humidity), these formulas become:

$$H_s = 1.06\,Q\,(t_o - t_i) \tag{9a}$$
$$H_l = 0.66\,Q\,(\omega_o - \omega_i) \tag{10a}$$

The various terms involved in these two formulas are easily determined. The method of determining the outdoor and indoor dry-bulb temperatures and vapor densities has been described previously under Design Conditions. The quantity of ventilation air, Q, is equal to the specified amount of ventilation air if a ventilation duct leads to the air conditioner. If dependence is placed on natural infiltration to supply outdoor air, Q is the infiltration rate.

Heat from People

The heat given off by persons varies with the degree of activity of the individual and the temperature of the air, as described in Chapter 2 of THE A. S. H. V. E. GUIDE.[16] In most air conditioning applications people are found in one of two general states of activity—either they are seated or standing at rest, moving about only occasionally (as in a home or office); or they are engaged in brisk activity such as walking or dancing (for example, restaurant waiters or night club patrons). For all ordinary cases, the values of sensible and latent heat gain in Btu per hour per person as given in Table 7 may be used. Consideration should, of course, be given in all cases to the number of people who fall in one category or another.

DETERMINATION OF TOTAL HEAT GAIN

Selecting the Time of Day for Calculation

In selecting the time of day for calculation, one must distinguish between two general types of application. The first class includes all applications which

[16] Loc. Cit. See Note 3.

do not have an unusually large and sharply variable heat gain from people or appliances, such as residences, offices, some types of retail shops, etc. The second class would include all applications which do have a large heat gain from people and appliances, varying sharply at certain hours of the day, such as restaurants, night clubs, theatres, funeral parlors, etc.

Cases in the first category may be handled quite simply and with reasonable accuracy. The time of maximum heat gain in such cases is dependent upon the climatic factors and degree of exposure of the space being air conditioned. It does not depend so much on variables like occupancy and heat producing appliances. Because of this it has been possible to make a study of the probable time of maximum heat gain in such cases with various exposures, and reach substantially accurate conclusions beforehand. The results of this study are given in Fig. 6 and Table 8. It is necessary merely to select from Fig. 6 that room which has the same exposure as the room being calculated, and then refer to Table 8 to find the correct time of day for that room.

It is not intended that Fig. 6 and Table 8 should be used blindly. This standard method was worked up on the basis that the rooms under consideration

TABLE 7. HEAT FROM PEOPLE (AT 80 F)

1. Average condition (at rest, eating, office work, etc.)		
Sensible heat	220	Btu per hour per person
Latent heat	180	" " " " "
Total	400	" " " " "
2. Medium rate of exertion (restaurant waiters, average dancing, etc.)		
Sensible heat	220	Btu per hour per person
Latent heat	420	" " " " "
Total	640	" " " " "

were approximately square; that outside walls were exposed to the sun, and not shaded by other buildings; that outside walls had one or more windows; that the heat from occupants and appliances was small as compared with the heat from other sources. Consequently, these data do not apply exactly to rooms which do not conform to these conditions, although it is quite correct for rooms which do not deviate too widely from these specifications.

As an example of the use of Fig. 6 and Table 8, consider a room which has a south and west exposure, no awnings on the windows, and an occupied space above, such as might occur in a modern office building. The sun effect on the south wall and windows reaches a maximum at noon. Sun effect on the west wall and windows reaches a maximum at 4 p.m., and the value thereof is greater than for the south wall. The outdoor temperature, and consequently the tendency to gain heat by conduction through practically all walls, ceilings, floor, etc., is greatest about 3 p.m. The heat from outdoor air also reaches a maximum about 3 p.m. From this it is apparent that the effects at 3 p.m., or thereabouts, will overshadow the effects at noon. Furthermore the sun effect on the west wall at 3 p.m. will be almost as great as at 4 p.m. Con-

sequently 3 p.m. should be the proper time at which to make the calculation. It will be noted that room 13 in Fig. 6 fulfills the specifications laid down above, and from Table 8 the time of maximum heat gain is listed as 3 p.m.

Now suppose that this same room were located under a roof or a hot attic. The sun effect above the room would be a large factor and would reach a maximum at noon, just the same as on the south wall. The effect of this would be to make the maximum total heat gain come somewhat earlier than before—probably about 2 p.m. instead of 3 p.m. This is shown in Table 8.

Let us suppose now that there is under consideration a restaurant doing its greatest business at dinner, and therefore having its maximum occupancy between the hours of 6 and 8 p.m. The people would naturally supply a considerable proportion of the total heat gain and would probably control the time of day when the total heat gain is maximum. Hence, it is not correct to make the calculations for 2 or 3 p.m., and then add the heat gain from occupants

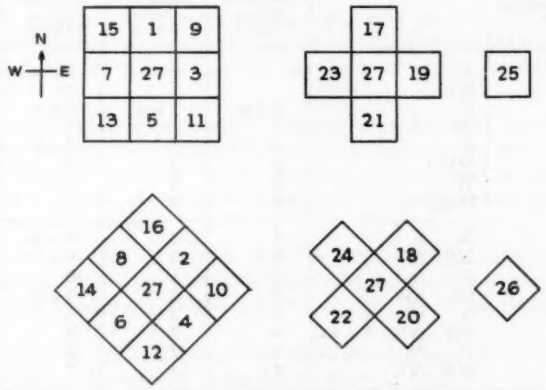


FIG. 6. TYPICAL ROOM EXPOSURES (USE WITH TABLE 8 TO DETERMINE TIME OF MAXIMUM COOLING LOAD)

which occurs at a later hour. On the other hand, it would not be quite accurate to make calculations of all components for 6 or 7 p.m., because of a factor which has been mentioned previously, but not discussed, namely, the effect of time lag or retardation of the heat flow through walls. This factor will be discussed in more detail later. The correct solution in the case of a restaurant of this nature would be to calculate the heat gain from sun effect through windows, lights and appliances, ventilation air and occupants at 6 or 7 p.m. (of course, in the particular case being discussed there would be no sun effect through windows at this late hour), and to add to them the conduction through walls and sun effect through walls calculated for about 4 p.m. or earlier.

In case of doubt as to the proper time to use, the following procedure is recommended:

1. Calculate the maximum value which each component attains during the day, assuming that there is no time lag in the heat transmission through walls.

2. Note the hours at which these peaks occur.
3. Consideration should be given to time lag phenomena in walls, in a manner to be discussed later.
4. Select the probable time of maximum total heat gain from an analysis of the data.
5. Where the economics of the application warrants the additional time required, an hour by hour calculation of the heat gain may be made.

TABLE 8. TIME OF MAXIMUM COOLING LOAD FOR ROOMS WITH DIFFERENT EXPOSURES

(To be used with Fig. 6)

ROOM NUMBER FROM FIG. 8	NUMBER OF WALLS EXPOSED	EXPOSED WALLS	AWNINGS ON WINDOWS		NO AWNINGS ON WINDOWS	
			OCCUPIED SPACE ABOVE	ROOF OR ATTIC ABOVE	OCCUPIED SPACE ABOVE	ROOF OR ATTIC ABOVE
1	1	N			2 P.M.	2 P.M.
2		NE			2	2
3		E	2 P.M.	2 P.M.	9 A.M.	9 A.M.
4		SE	1	1	10	10
5		S	2	1	1 P.M.	1 P.M.
6		SW	3	2	4	3
7		W	3	3	4	4
8		NW	4	3	5	4
9	2	N E	2	2	9 A.M.	9 A.M.
10		NE SE	2	1	9	9
11		E S	2	1	10	10
12		SE SW	3	2	3 P.M.	3 P.M.
13		S W	3	2	3	3
14		SW NW	3	3	4	4
15		W N	3	3	4	4
16		NW NE	4	3	5	5
17	3	W N E	4	3	4	4
18		NW NE SE	3	3	4	4
19		N E S	2	2	10 A.M.	10 A.M.
20		NE SE SW	3	2	3 P.M.	3 P.M.
21		E S W	3	2	3	3
22		SE SW NW	3	3	4	4
23		S W N	3	3	4	4
24		SW NW NE	4	3	4	4
25	4	S W N E	3	2	3	3
26		SW NW NE SE	3	2	4	4
27	None				2	2

The Effect of Heat Storage in the Walls

Very little has been said thus far about the subject of heat storage in walls, yet it would seem that it must certainly have some effect on the heat gain of a structure. What, then, is the justification for ignoring it in the method of calculation which has been described? The answer is that this effect has not been ignored, but its discussion has been deferred in order to not complicate

the picture unduly. Actually, in a great many cases the effect of heat storage in walls may be ignored without creating a serious error in the calculations. Whether this effect may or may not be ignored depends largely on the relationship between the magnitudes of the conduction and sun effect through walls and the other components of the heat gain. If the two former components are predominant, then it may be desirable to take account of the effect of heat storage. On the other hand, if these two components are small as compared with the rest of the heat gain, the neglect of heat storage phenomena will produce no serious error.

The method of calculating conduction and sun effect through walls, described, has been compared with some very accurate tests [17] conducted a few years ago on walls having various heat capacities, and has been shown to give highly accurate results for walls which have heat capacities comparable to those found in ordinary building construction. For walls having abnormally high capacity the calculations give results which are slightly on the conservative side. This comparison demonstrated that the heat capacity of normal walls has relatively little effect on the amount of heat transmitted through them, but it does give rise to a time lag of one hour or more, depending on the construction.

Until more complete experimental evidence is available on the difference of time existing between the manifestation of maximum outdoor effects and maximum heat gain from walls of various building materials, and on the amount of reduction in total heat conducted on account of the storage effect, the following procedure is recommended:

1. Calculate the heat from sun effect through windows, appliances, lights, ventilation air and people on the basis of temperatures, sun effect factors, and other conditions actually existing at the time of maximum total heat gain.
2. Add to the above conduction through walls and sun effect through walls calculated on the basis of temperatures existing from 3 to 6 or more hours earlier in the day than the time at which the maximum total heat gain occurs. Reference may also be made to Chapter 8 of THE A. S. H. V. E. GUIDE [18] for time lag factors.

Unusual Sources of Heat Gain

The obvious sources of heat which have been discussed are sometimes supplemented by hidden and unexpected sources, the possibilities of which must not be overlooked. One fairly obvious one is the conduction of heat through the walls of the ducts in a central plant system, particularly where the ducts run through unventilated attics, outside walls, etc. The amount of heat thus conducted to the conditioned air can be calculated from a knowledge of the temperatures inside and outside the duct, and from a consideration of the usual heat transfer phenomena. In general, the following formula may be used with sufficient accuracy.

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[17] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 231.

[18] Loc. Cit. See Note 3.

$$H_d = A_d U_d (t_1 - t_2) \tag{11}$$

$$U_d = \frac{1}{\frac{1}{1.2} + \frac{t}{k}} \tag{12}$$

where

H_d = heat transmitted through walls of duct system, Btu per hour.
A_d = area of duct surface, sq ft.
t_1 = temperature of air outside the duct, F.
t_2 = temperature of air inside the duct, F.
t = thickness of insulation applied to duct, inches.
k = conductivity of duct insulation, Btu per hour per sq ft per F per inch thickness.

Another unusual source of heat gain arises when rooms or furred spaces are used as plenum or return air chambers. Sometimes the walls of such chambers adjoin very hot spaces, like kitchens or attics, and sometimes they are exposed to the outside. When such cases are encountered, it is necessary to calculate the heat gain of the air chamber as well as the heat gain of the air conditioned space.

A third source of unusual heat gain arises when steam pipes are hidden in the walls and carry steam during the summer time. This might happen in hospitals or various industrial plants, for example. Such cases as this are rare but have been known to result in the maintaining of unsatisfactory indoor conditions when their existence was not known before the equipment was installed. Calculation of heat gain from such sources may be made on the basis of the temperatures involved and the usual heat transfer phenomena.

Methods for Reducing Heat Gain

The application engineer should always be on the alert for opportunities to reduce the heat gain of a structure where means are available to do so in such a manner as to reduce the net cost of the installation over the period of its life.

The use of insulating materials in walls which have a heavy heat gain is one obvious method of reducing the size of equipment required. The use of awnings, blinds or shades is an aid in the reduction of sun effect through windows. Of course, such shades, blinds, etc. are of no benefit unless they are drawn. The spraying of water on roofs which are directly over the air conditioned spaces is another method which has been tried successfully at times. This water may sometimes be taken from the water discharge circuits of the condensing unit or from an independent circuit. Attic ventilation is another means of effectively reducing the heat gain through the roof. In many cases, unventilated attics situated beneath roofs which are exposed to the full intensity of the sun may rise to temperatures as high as 130 or 140 F. Since the outdoor air is seldom hotter than 110 deg, it has the capacity to remove heat from such attics, provided it is forced through by a suitable fan or blower. The fan should be able to provide in the neighborhood of 30 complete air changes per hour in the attic space. Under such conditions the average air temperature in the attic space would usually be from 5 to 10 deg warmer than outdoors during the middle of the afternoon, when the sun effect is greatest. There are times, of course, when attic insulation may be more valuable than attic ventilation.

SUMMARY AND CONCLUSIONS

This paper has described a complete and systematic method for determining the character and magnitude of the cooling load for air conditioning systems. The salient features of this method may be summarized as:

1. It organizes and systematizes the work so that a complete and accurate calculation requires a minimum of both time and technical experience.
2. A new and simple method is presented for accurately calculating the sun effect through opaque walls and roofs.
3. The method of separating the components of the total load is such that the total sensible and total latent heat gains may be determined for any hour of the day.
4. An approximate method of allowing for time lag effects resulting from heat storage further improves the accuracy of the calculations.
5. A procedure is given for predetermining the time of day when the total heat gain is greatest.
6. The general procedure is applicable to an hour-by-hour calculation of the cooling load, if the economics of the application warrants it.
7. Although the discussion has been centered around the determination of the maximum total heat gain, it is equally adaptable to the determination for various conditions of operation, and may be used to determine the maximum ratio of dehumidification to sensible cooling.

The method of making heat gain calculations described herein has been under development for several years, during which time it has undergone many refinements of detail. However, it has been used in fundamentally the same form as presented here for over 2 years by a large number of application and sales engineers. The experience gained from the many installations made on the basis of cooling loads calculated in this manner has substantiated its simplicity and reliability.

The method is not perfect. There is a large field for improvements. Some of the points which need elaboration and refinement are:

1. The effects of heat storage in walls on the cooling load when a constant indoor temperature is maintained.
2. The effects of heat storage on the pull-down load after the cooling system has been shut down for some time, or following a sharp change in outdoor temperature.
3. More complete data on infiltration rates during the cooling season.
4. More complete data on the load reducing properties of various types of shading devices, expressed as a function of the type of material and age.
5. More complete and accurate data on the ability of various outside building surfaces to absorb solar radiation, expressed in terms of the type and age of the surface, and the angle of the surface with respect to the rays of the sun.
6. A more complete analysis of climatic data.

It is hoped that further research will provide the information necessary for making additional refinements in the cooling load calculations.

DISCUSSION

W. H. CARRIER: In general I think this article is entirely competent and indicates clear and constructive thinking. I should list the following data and ideas as being

valuable inasmuch as they are either new or have been insufficiently stressed in past practice:

1. The method of determining sun effect through walls and roof. I quote, "Fundamentally, this method is equivalent to calculating the temperature differences across walls exposed to the sun, and avoids the necessity for estimating them. The mathematical elimination of the wall surface temperature, however, results in the simple expressions given by equations. . . ." I think as far as it goes this method has a great deal to commend it. I have checked their radiation factors F and they are based on a combined outside convection and radiation and transmission coefficient of between 4 and 4.3, which appears reasonable. The data in Table 3, Sun Effect Coefficients at Various Latitudes for Walls Facing Several Directions, are quite complete.

2. The method of selecting the time of day when the total heat gain is greatest for various exposures, shown in Fig. 6 and Table 8 is useful if not taken too literally.

3. The importance of mean daily temperature range and its variation with locality, shown in Fig. 2 and Table 1, is worthwhile and has not been sufficiently stressed in the past.

The following criticisms of the paper can be made:

1. The curves in Fig. 1 are misleading because lag and storage have not been taken into account. While this is fully explained further on in the paper, a false impression is created in the mind of the reader at this point.

2. There is an obvious error in the fourth paragraph under the side heading Design Conditions, where 90 per cent should, of course, read 10 per cent.

3. The paper cites THE A. S. H. V. E. GUIDE 1935 as the source for design outside air temperatures and transmission factors. I am questioning both and do not feel they are correct.

4. It seems difficult to subscribe to the formulae in Table 2 for unventilated attics, since it appears to me that the re-radiation effect between the roof and the ceiling, which accounts for virtually all the heat transmitted in the summer time, is not rationally expressed.

5. I think the weakest part of the article is on the subject of heat storage. The tendency here is to neglect the effect of storage completely on the assumption that it usually can be ignored. Inspection of the data in the reference which they cite in Note 10 is sufficient refutation of this fact. I have checked the values which their equation FaI gives for several wall and roof constructions. For roofs with a transmission coefficient of between 0.25 and 0.30, a medium absorption and a normal sun effect coefficient for 40 deg latitude, the additional equivalent temperature differential added to the normal differential between outside and inside is about 45 deg, and this is entirely independent of the mass or material used. We at present figure differentials of 60, 45 and 30 added to the normal temperature head for light, medium and heavy roofs respectively. The sun effect on walls gives an equally misleading result, in this case 20 to 30 F over the normal differential, which for a wall of any mass at all is entirely too high. The criticism then is not of the method used, which I think is inherently sound, and has much to recommend it as rational, but that the method does not go far enough to account for storage, which according to all the evidence I am able to adduce, is a highly important factor.

M. K. FAHNESTOCK: Referring to Fig. 5 of the paper, showing the solar radiation factor vs overall wall transmission coefficient, I would like to ask the authors if this curve does not shift up or down depending upon the difference between the temperature of the outside air and the temperature of the outer surface of the wall.

I would also like to ask if items 1 and 2 in the summary on pages 341 and 342 are based upon actual experimental data or are they merely statements of the authors based upon practical field experience?

AUTHORS' CLOSURE: The authors wish to take this opportunity to express their appreciation for Mr. Carrier's discussion. His criticisms are well taken, as they emphasize several points in the paper which undoubtedly should have been discussed more fully. The following comments on Mr. Carrier's discussion are made in order:

1. A note has been added to the title of Fig. 1 to the effect that the curves neglect the effect of time lag, in order to avoid the possible misunderstanding pointed out by Mr. Carrier. The curves in Fig. 1 were presented for the purpose of illustrating how variations in the several components of the cooling load result from variations in different design conditions. Since the design conditions do not all vary in the same manner, the shapes of the load curves are different, and the maximum values do not necessarily occur simultaneously. It was felt advisable to avoid injecting the further complicating effect of time lag at this point of the discussion, as an understanding of the dependence of load variation upon variation in design conditions is essential to a rational study of the effect of time lag.

2. In the fourth paragraph under the side heading Design Conditions, 90 per cent was in error. This has been corrected to 10 per cent in the final proof.

3. The authors did not intend to create the impression that they were recommending any particular source of numerical data for design temperatures or heat transmission coefficients. As a matter of fact, values which they use in practical calculations deviate, in some cases, from those found in the several sources cited in the paper. The paper was intended as a discussion of a *method* of heat gain calculation rather than a complete working reference to recommended numerical data.

4. The formulas for unventilated attics, given in Table 2, are based on the assumption that both radiation and convection are effective in transferring heat from the roof to the ceiling. Actually, only radiation is effective, because the flow of heat is downward, and hence free convection is suppressed. However, corrections for this factor were omitted in order to avoid further complication, because in general the net error is small. The formulas in Table 2 in effect assume two film resistances in series, one from the ceiling to the attic space and one from the roof to the attic space. These film resistances include the effects of both radiation and convection, which are both of the same order of magnitude. Actually, there is only one resistance between the roof and ceiling—radiation. Hence, while the formulas in Table 2 are not rationally expressed, they afford a convenient means for arriving at a practical result.

5. The effect of time lag resulting from heat storage is sometimes an important one, as noted by Mr. Carrier. The authors have shown how this effect can be approximated under the side heading The Effect of Heat Storage in Walls. The method presented there affords a rational attack to the problem, and is applicable to any combination of circumstances. At the present time, accurate estimates of the effects of heat storage are difficult because of the lack of experimental data. As pointed out in the paper, more complete information is required concerning:

(a) The effect of heat storage capacity on the time displacement of the curve of wall conduction with respect to the curves of temperature difference and intensity of incident solar radiation.

(b) The effect of heat storage capacity on the magnitude of the actual transient heat flow with respect to the magnitude of an assumed steady-state flow.

In reply to Professor Fahnestock's first question, the curve in Fig. 5 does shift up or down slightly with various temperature conditions. This curve is approximately U_w/f_o. Hence, any factor which changes the value of f_o affects the value of F for any value of U_w. The coefficient f_o includes both convection and radiation. The convection component varies with wind velocity, nature of the surface and somewhat with temperature difference between the outside surface of the wall and the surrounding air. The radiation component of f_o varies with the absolute temperature of both the outside wall surface and the outside air. The authors are of the opinion

that for practical calculations corrections to the value of F for variations in f_o resulting from variations in the general temperature level of heat transfer by radiation, or resulting from the effect of the roughness of the surface upon the convection component are not necessary.

In reply to Professor Fahnestock's second question, the statement made under item 1 in the summary on page 341 is based on both practical experience in the field and experimental data. Experimental studies of the effects of various types of shading appurtenances have been previously reported.[19] These data require careful analysis, however, in order to segregate the effects of conduction and direct solar radiation. The statement under item 2 of the summary on page 342 is based entirely upon field experience.

[19] Loc. Cit. See Note 7.

No. 1022

OIL BURNING IN RESIDENCES

By D. W. NELSON * (*MEMBER*)
MADISON, WIS.

OIL burning is a very desirable means of heating residences, because of its automatic temperature control and the almost complete lack of attention required to the heating plant throughout the heating season. In order that oil burning may hold its competitive position in the future with other methods of heating, it will become increasingly necessary to eliminate losses and inefficiencies that result in excessive heating costs.

In the past it is likely that too many installations have been made in a haphazard way. Too often the salesman has been only interested in the placement of the burner and has entirely ignored probable costs of heating. Many burners have been installed in under-sized or inefficient boilers or furnaces. In some cases the combustion chamber design has been incorrect and in others the rate of burning has been much higher than required by the heat losses of the residence.

The study in the present paper was made to investigate at least one of the losses and to make a survey of actual installations in residences as to the efficiency of oil burning.

This report is divided into two parts. The first concerns an analysis of losses occurring during off-periods in the operation of intermittent types of burner. The second concerns the results of a survey of a large number of residence installations as to efficiency of combustion as determined by CO_2 and flue gas temperature readings and other related items. The work was done under a project [1] of the Civil Works Administration and later had the assistance of students enrolled under the Federal Emergency Relief Administration.

Most burners work on the intermittent principle. When the call for heat is made the burner starts, and the combustion improves as temperatures increase in the brick work of the combustion space. The burner runs varying lengths of time and makes a varying number of starts depending upon the heat requirements of the building at that time. The newer temperature control systems give very close regulation of temperatures, and the running periods are conse-

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* Asst. Professor of Steam and Gas Engrg., University of Wisconsin.

[1] The following worked on this project under CWA: W. F. Bakke, E. P. Hansen, L. E. Heger, F. C. Rossow; and the following under FERA: R. H. Amundson, G. W. Androne, H. C. Mittelstaedt, R. H. Newbury, and R. A. Wilson. Project carried on in the Dept. of Mech. Engrg., University of Wisconsin, Prof. G. L. Larson, Chairman.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June, 1935, by D. W. Nelson.

quently shorter and more frequent. This tends to increase the off-period losses which result from the cooling off of brick work and heating surfaces with the passage of air through the heating unit while the burner is shut down. Most burners operating on the high-low flame principle are subject to a somewhat similar loss due to a lowered temperature of the combustion space and a larger percentage of excess air during the low flame periods.

LABORATORY INVESTIGATION

The present study was made on a pressure atomizing burner with positive or fan air supply. The tests were made on a laboratory installation completely

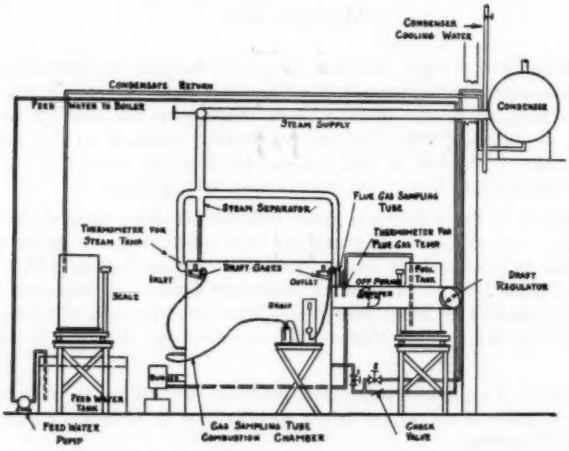

FIG. 1. ARRANGEMENT OF LABORATORY EQUIPMENT TO DETERMINE OFF-PERIOD LOSSES

equipped for testing and on a residence installation under actual conditions of heating.

Fig. 1 shows a diagrammatic sketch of the laboratory test equipment. The boiler used was a 10-section rectangular cast iron water tube steam boiler having about 87 sq ft of heating surface, and was a coal burning type often used for oil burning. With a 2½ gal nozzle, the maximum load developed was about 850 sq ft of radiation. The net combustion space for oil burning was 8 cu ft. For the purpose of these tests a closely fitting damper was placed in the smoke pipe near the boiler outlet. The tests were made to determine the difference in efficiency with various on and off periods for operation with and without the damper. When the damper was kept in the open position at all times, the operation corresponded to the usual method of operation with this type of burner. This was compared with operation when the damper was closed manually as soon as the burner stopped and was opened 1 min before the burner started. With each of the two types of operation various percentages of off and on time were tried. The on periods were 10 min and the off

periods were of varying length. For instance at 50 per cent operating time the burner would be on 10 min and off 10 min and at 25 per cent operation it would be on 10 min and off 30 min.

The CO_2 analysis was maintained at 10 per cent with No. 2 distillate fuel burned at the rate of 2½ gal per hour. The flue gas temperature measured about 450 F. A draft control was used and an effort was made to maintain 0.02 in. in the combustion space. Due to the windy weather at the time of some of the tests and to an oversize chimney, difficulty was experienced in holding this steady and as a consequence it was found difficult to always get consistent results. A draft regulator was used at the location shown in Fig. 1. The

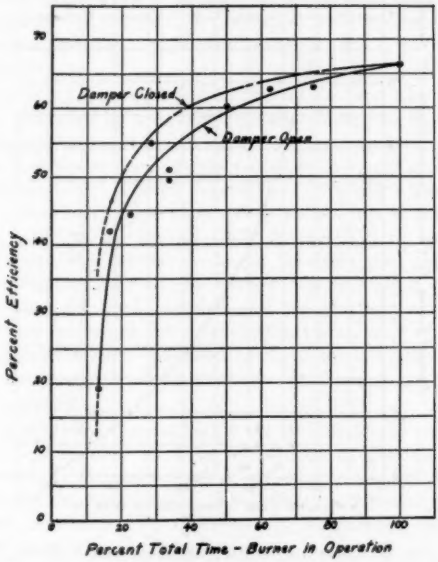

FIG. 2. COMPARISON OF EFFICIENCY WITH AND WITHOUT THE OFF-PERIOD DAMPER

results of the tests are shown on Fig. 2 for both types of operation. The maximum efficiency secured at continuous operation was 66 per cent. The CO_2 determination of 10 per cent and the boiler outlet temperature of 450 F indicate an efficiency of 82.5 per cent. The reason for this difference of 16.5 per cent is not known. The stack gases were not analyzed for unconsumed hydrogen or hydrocarbons. It is suspected that the heat loss to the base and floor was large, as the combustion space was partly below the water-cooled surface of the boiler. This arrangement was made because of a drop section in the boiler useful in connection with the hand firing of bituminous coal. This loss might have been reduced by the use of insulation between the bricks and concrete base and floor or by the use of insulating bricks.

With the interpretation of the test results as made by the placement of the

two curves of Fig. 2, the saving due to the use of the damper appears as the vertical distance between the two curves. These differences plotted on Fig. 3 show the greatest saving for the damper for the shortest periods of operation. It would seem for best efficiency that the burner should operate almost continuously under maximum heat requirements. In this locality the average load is about one-third the maximum. This indicates an average loss of about 9

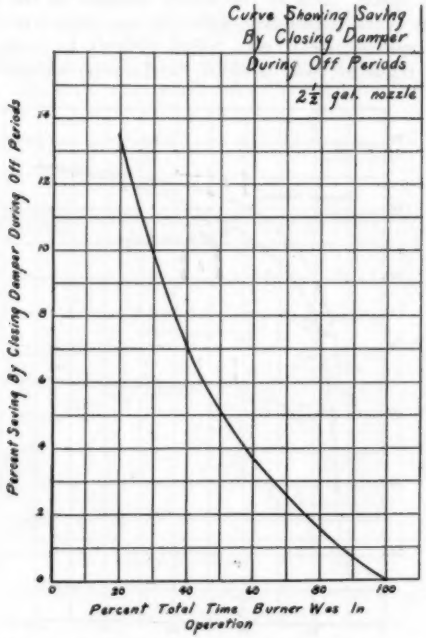

FIG. 3. SAVINGS RESULTING FROM THE USE OF THE OFF-PERIOD DAMPER AT VARIOUS PERCENTAGES OF RUNNING TIME

per cent that might be eliminated by the prevention of air circulation through the combustion space during off-periods.

The second part of this study of the off-period loss was made in a residence heated by a pressure atomizing burner installed in a rectangular steam boiler of 8 sections. The system was a one pipe system with vacuum vents and the connected load was about 350 sq ft EDR. The boiler catalog rating for coal burning was 900 sq ft and the net rating would be about 450 sq ft. The rate of oil burning was 1.8 gal per hour which resulted in a boiler stack temperature from about 500 F to 550 F. The draft regulator setting was kept constant on all tests. A fuel burning rate of 1.8 gal per hour gives ample capacity for the warming up period in the morning when operated on dual (lower night temperatures) control. The system operates on pressure control for several hours

during the warming up period in cold weather indicating sufficient burner and boiler capacity, but this does indicate insufficient radiation for this type of operation. Heat loss calculations indicate the installed radiation as being sufficient to meet the maximum losses but there is no reserve for rapid heating up after the temperature has been allowed to drop to 50 F. One downstairs radiator was not used during these tests.

The use of oil was measured by totalling the hours of burner operation by a self-starting electric clock placed in the burner motor circuit so that it started and stopped with the burner. Since the oil temperature was practically constant because an inside oil tank was used and also since the same nozzle and the same atomizing pressure was used on all tests, the running times gave an accurate determination of oil use. An accurate fluid column gage on the tank was used as a check and the rate of consumption of 1.8 gal per hour was obtained by this means.

An automatic damper was installed in the smoke pipe near the boiler outlet. The damper was operated by an electric motor similar to that used for operation of draft dampers on hand-fired installations arranged for thermostatic control. The room thermostat caused this motor to open or close the damper to suit the demands for heat. The oil burner was controlled through this damper by switches of the mercury type attached to its shaft. Whenever the damper was in the open position, the burner controls caused it to start and when the damper motor caused the damper to begin to close, the controls stopped the burner. The cycling controls of the burner allowed about a minute between the damper assuming the open position and the starting of the burner. The early breaking of the burner motor circuit upon the turning of the damper from the open position and the slow operation of the damper motor gave considerable time for burnt gases to be cleared from the combustion chamber at the close of a period of operation.

Tests with one system of operation were usually made over a period of a week and a sufficient number of days were taken to eliminate to some extent the variable factors of sun and wind effect which in addition to temperature differences influence the heat losses. The effect of the damper on the amount of fuel used was observed with continuously maintained inside temperatures and with dual or lowered night temperatures.

The results are shown in Fig. 4 for a continuously maintained temperature of 70 F and in Fig. 5 for daytime 70 F and night thermostat setting of 50 F. The outside temperatures, wind velocities and sun intensities shown on these figures are averages of readings secured from the local Weather Bureau Station.

The data on Fig. 4 show that the weather conditions were quite similar in the two periods of tests for continuous control. During the 18 days of tests with the damper in use, the temperature averaged 24 F, the wind velocity 10 mph and the sun effect 206 cal per sq cm. During the 13-day period without the damper in use, the temperature averaged 1 deg lower, the wind velocity the same, and the sun intensity 99 cal per sq cm. The running time with the damper in use was 5.8 hours per day and without the damper was 6.5 hours per day. When the correction is made for the 1 deg difference in temperature, the saving is 8.9 per cent. This saving in running time shows as the vertical distance between the two curves on Fig. 4 at 23 F. The sun intensity variation

favored the period when the damper was in use which indicates that the real saving was somewhat less than 8.9 per cent. The scattering of the points of

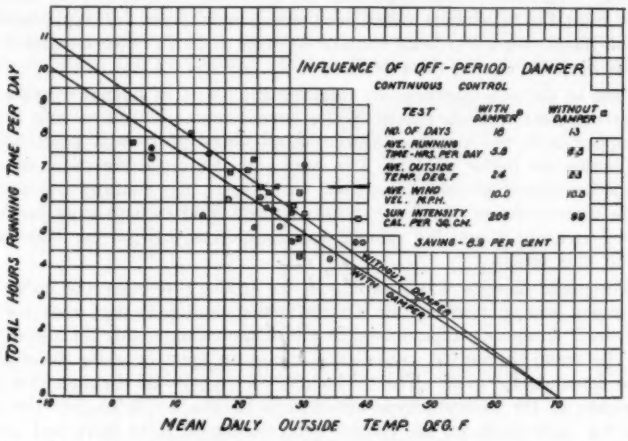

FIG. 4. RESULTS OF OFF-PERIOD DAMPER TESTS IN RESIDENCE WITH CONTINUOUSLY MAINTAINED TEMPERATURE OF 70 F

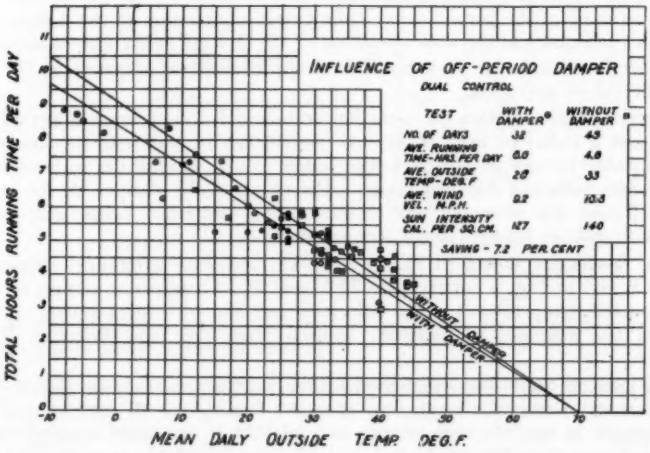

FIG. 5. RESULTS OF OFF-PERIOD DAMPER TESTS IN RESIDENCE WITH LOWERED NIGHT TEMPERATURES

daily average temperature and the hours of burner operation can be explained when the sun intensities and wind velocities of individual days are examined. A point above the line is almost always a windier-than-average day and a point below is a sunnier-than-average day.

The similar comparison of damper operation when carrying reduced temperatures at night time from 10:30 p.m. to 5:45 a.m. indicated a saving of 7.2 per cent as shown on Fig. 5. In these tests of 32 days with the damper and 45 days without the damper, the sun intensities and wind velocities varied a slight amount, but in directions tending to balance each other so that probably these factors have not appreciably influenced the results. The temperature averages differed by 13 F, being 20 F for the period with damper use and 33 F without the damper. The carrying of lowered temperatures at night times does make a saving which differs at various outside temperatures. This influences the results of this damper study since the average outside temperature differed by 13 F in the two periods. The saving due to such dual control must result from smaller temperature differences. In this study of an off-period damper, the average running time of 6.0 hours at 20 F was corrected to 33 F by a ratio of temperature differences inside to outside with 20 F weather to temperature differences inside to outside with 33 F weather. In order to obtain these differences the inside room temperatures were averaged by hours for each condition from recording charts. The curves on Fig. 5 as on Fig. 4 were drawn as straight lines through 70 F and a point established by the average running hours and the average outside temperature. The saving is approximately the vertical difference between the two curves at 33 F.

The saving found by the use of the damper was 7.2 per cent for dual temperature control and 8.9 per cent for continuous control. The latter figure is considered somewhat high because of differences in sun effects and the 7.2 per cent is considered somewhat low because of the method of damper operation.

The room thermostat controlled the operation of the damper. During the warming up period each morning, there was a time when the burner was controlled by the steam pressure limiting device which allowed the steam pressure to vary between ½ lb per sq in. and 4 lb per sq in. The colder the weather the longer the period of such pressure controlled operation and during all of this period the damper remained open continuously. Since the starts and stops are quite frequent and the brick work would be at a red heat, the loss in off periods during these warming up times would be appreciable. Therefore the figure of 7.2 per cent saving for the damper during the dual operation period is considered to be somewhat low.

Fig. 6 shows a chart of flue gas and boiler water temperatures for a typical day with damper operation and Fig. 7 is a similar chart for operation without the damper. The boiler water dropped in temperature to about 160 F during the night period of lower room temperatures when the off-period damper was in use as shown on Fig. 6. Without the use of the damper the water temperature dropped to about 140 F. These values were checked by mercury thermometers. The heat loss involved in this 20 deg difference is not appreciable and cannot account for the saving secured with the damper. The charts show a period of 15 min to bring the water to boiling during the warming up period on days when the damper is used and about 5 min longer on days when it is not used. The efficiency during this period would be low due to cool brick work and to the necessity of bringing it up to temperature. The water temperature record pen led the flue gas temperature pen on this recording instrument by 15 min, to allow for the passing of the pens. The flue gas temperatures were checked

with a mercury thermometer at the upper range, but are known to be too high at the lower range. It is believed that the flue gas temperature approaches very closely the water temperature during the long off period at night. When the off-period damper is used, air circulation past the thermometer bulb is stopped, which affects the reading. The cooling of the boiler water after periods of operation under control of the thermostat is due not only to boiler losses, but

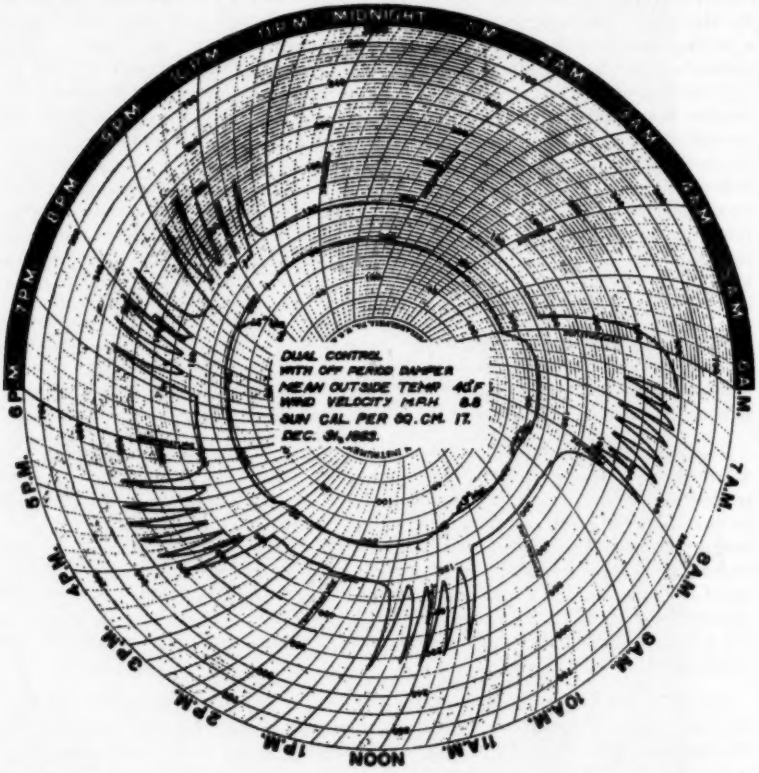

FIG. 6. TYPICAL DAILY CHART SHOWING FLUE GAS AND BOILER WATER TEMPERATURES WITH OFF-PERIOD DAMPER IN USE

also to the generation of steam at sub-atmospheric pressures. As shown on Fig. 6 a vacuum of 10 in. mercury existed at 9 a.m. after the period of pressure controlled operation and 2 hours later when the thermostat called for additional heat, there still was a 3 in. vacuum. Since all of the material in the house has not been brought up to temperature in the warming up period of pressure control, the vacuum secured directly after this period is greater than after periods under control of the thermostat. The periods of operation under control of the thermostat, which has a built-in heating coil, were found to be all of about the

same length and averaged about 6 min. During this 6 min period, the flue gas temperature rises. During the long initial running period early in the morning, the flue gas temperature approaches a stable value such as would be obtained on continuous operation. Fig. 6 shows 26 starts on that day and Fig. 7 shows 43 per day. The greater number in the latter case seems to be due to the greater wind velocity. On continuously maintained temperature operation,

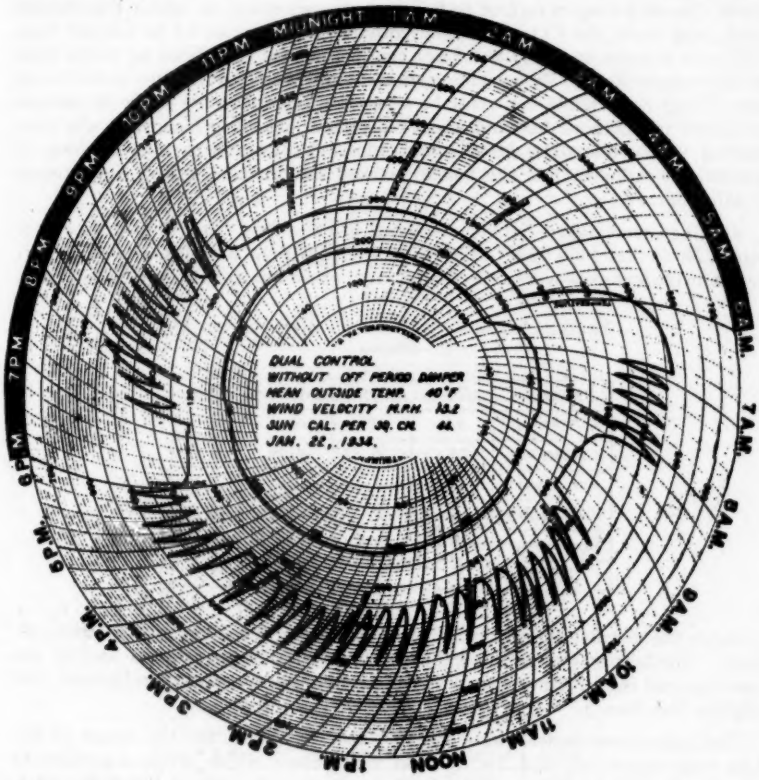

FIG. 7. TYPICAL DAILY CHART SHOWING FLUE GAS AND BOILER WATER TEMPERATURES WITHOUT THE USE OF THE OFF-PERIOD DAMPER

the number of starts would be greater. After each period of operation an off period follows in which losses occur, part of which may be prevented by shutting off air circulation through the boiler at such times.

FIELD SURVEY

In the laboratory it has not been found difficult to make an installation that would result in 10 per cent CO_2 with clean burning on intermittent operation.

It was possible to reduce the excess air on continuous operation and obtain considerably higher CO_2 than 10 per cent, but these same adjustments could not be maintained during intermittent operation because of the less favorable temperatures of the brick work and consequent smoking.

Previous miscellaneous tests in residence installations had indicated a wide variation in the efficiencies of combustion and heat absorption. As an example, when the oil burner was first installed in the residence on which the damper study was made, the CO_2 obtainable without smoking was 6.0 to 6.5 per cent. This was brought up to 8.0 to 8.5 per cent by various alterations up to the time of the damper study. By rebuilding of the combustion space, there is no reason why 10 per cent should not easily be secured. Since the tests made on various residence installations had indicated wide variations in efficiency, it was considered worthwhile to make a survey of as many residence installations as possible in order to find the variations and the average figures for present installations.

Readings were taken on 141 residence installations. Steam systems represented 78 of these, hot water systems 29, and warm air 34. Burners of 28 different manufacturers were represented some of whom had two types and Table

TABLE 1. TYPES OF BURNERS REPRESENTED IN SURVEY

Types	Steam	Hot Water	Warm Air	Total
H.P. Atom.	33	12	11	58
L.P. Atom.	17	5	4	26
Hor. Rotary	4	3	0	7
Vert. Rotary	15	7	6	28
Vaporizing	7	2	13	22
Totals	78	29	34	141

1 shows the distribution of burner types among the three kinds of heating systems. The largest number made by any one manufacturer was 21 and of any one type and manufacture was 19. The average number per manufacturer was slightly less than 5.

The main items desired were the CO_2 readings indicating the extent of the loss from excess air and the flue gas temperature which while dependent to some extent on the excess air is also dependent on the rate at which the oil is being burned and the ability of the heating surface to absorb that heat. These two readings give a means of finding the largest part of the losses in the form of heat passing up the chimney. It therefore becomes possible to determine the maximum possible efficiency of the installation. The actual efficiency is somewhat less because of possible escape of hydrogen and hydrocarbons as fuel items to the chimney and radiation losses to the basement and floor. Heat loss to the basement is useful provided the basement is not overheated for the purpose for which it is intended, and some heat loss to an inside chimney helps to supply useful heat to the house. It is therefore quite possible that the unmeasured losses are offset by unmeasured gains and that the efficiency as

determined by CO_2 and flue gas temperature at the boiler outlet is approximately the efficiency involved in the cost of heating.

The CO_2 determinations are shown in Table 2 tabulated as to systems and as to the number falling in each range of 1 per cent. The CO_2 was determined at the boiler outlet and in the combustion space whenever easily possible. In some installations, it was determined in only one of the two locations and the number of readings is given directly below the average in each column. The

TABLE 2. CO_2 DETERMINATIONS OBTAINED IN THE SURVEY

% CO_2	Steam		Hot Water		Warm Air		Total	
	Comb. Space	Smoke Pipe	Comb. Space	Smoke Pipe	Comb. Space	Smoke Pipe	Comb. Space	Smoke Pipe
Below 5		5				1		6
5-6	4	12		4	1	2	5	18
6-7	6	16	4	2	3	8	13	26
7-8	7	10	2	8	4	8	13	26
8-9	4	12	3	8	1	7	8	27
9-10	5	9	1	2	1	3	7	14
10-11	11	7	1	2	2	5	14	14
11-12	4	3	1	1			5	4
12-13	3	5		1			3	6
Above 13	3						3	
Ave	9.06	7.78	7.94	7.98	7.55	7.69	8.62	7.80
No	47	79	12	28	12	34	71	141

TABLE 3. FLUE GAS TEMPERATURE MEASUREMENTS OBTAINED IN THE SURVEY

Flue Gas Temp. °F	Steam	Hot Water	Warm Air	Total
Less Than 200				
200 - 300	1	1	2	4
300 - 400	4	1		5
400 - 500	7	4	2	13
500 - 600	12	5	6	23
600 - 700	24	9	9	42
700 - 800	29	9	15	53
Above 800	3			3
Ave. Temp °F.	642.6	612	634.8	634
Number	80	29	34	143

average CO_2 obtained in the combustion space was 8.6 per cent in 71 installations. The average at the smoke pipe connection was 7.8 per cent in 141 installations. The CO_2 content was determined in both locations in 57 cases. The average of the CO_2 in the combustion space was 9.1 per cent and at the smoke connection was 7.4 per cent. This difference, if real, indicates the infiltration of air between the two locations. It is, however, difficult to obtain an average sample in the combustion space since mixing or combustion may not be complete at the point of sampling. A 5/16 in. copper tube closed at the end and with small holes along the side was used for sampling.

Considering 10 per cent as a reasonable standard, 24 installations proved to be as good or better than might be expected. Six installations were operating with such a low excess of air that CO_2 was 12 to 13 per cent. As great a number as were above the dividing line of 10 per cent were below 6 per cent, which indicates in these cases the use of an average of about 200 per cent excess air. No attempt was made to change adjustment so that the findings represent actual residence heating conditions during average winter weather. In some cases only a change in air adjustment might improve the CO_2 determination, while in other cases the large amount of excess air is necessary due to faulty fuel preparation or lack of mixing of fuel and air at proper temperatures.

The flue gas temperatures found on the various systems are shown in Table 3. These temperatures were taken in the smoke pipe directly beyond the connection to boiler or furnace. The reading was taken after several minutes of burner operation and is considered to be close to the average for the usual

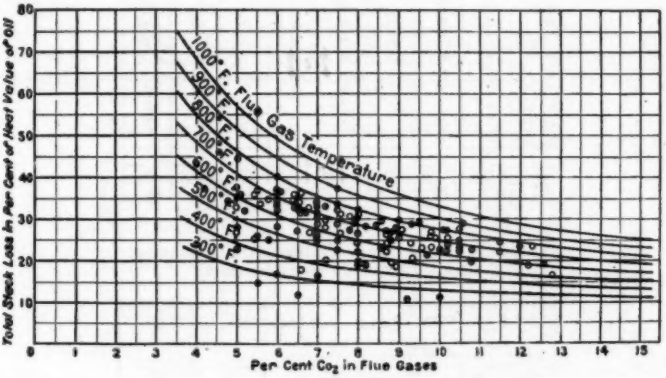

FIG. 8. RESULTS OF SURVEY AS TO FLUE GAS TEMPERATURE AND CO_2 SUPERIMPOSED ON A STACK LOSS DIAGRAM

running periods of burners. The average secured for all installations observed was 634 F. Considering 500 F as a reasonable standard, 121 installations showed a higher outlet temperature than this standard and 22 a lower value.

The survey results as to CO_2 and flue gas temperatures as given in Table 2 and 3 are shown graphically on Fig. 8. This curve sheet has been taken from a commonly quoted Government bulletin and shows total stack loss in per cent of heat value of the oil for various flue gas temperatures and CO_2 values. The survey data of individual determinations of CO_2 and flue gas temperatures have been superimposed on this as points. The average of 7.8 per cent CO_2 and 634 F indicates a stack loss of 27 per cent. The reasonable standard for an efficient installation, 10 per cent CO_2 and 500 F, indicates a loss of about 18.5 per cent. The probable maximum efficiency of the average installation observed would be 73 per cent. Other losses, such as unconsumed fuel items, radiation and conduction losses, would tend to make this lower and recovered heat from the smoke-pipe and chimney would tend to increase it. There is also a loss of heat during the off periods that must be made up by longer running periods,

which therefore causes a reduction in efficiency in the production of useful heat to the rooms.

The largest loss measured was with a CO_2 of 4.6 per cent and a flue gas temperature of 780 F and amounted to 46 per cent. The smallest loss amounted to 11 per cent and was found at temperatures of less than 300 F. It is interesting to note that provided the outlet temperature is low the loss of efficiency due to low CO_2 is not serious. Sufficient heat absorption surface for the burner capacity used seems most essential.

An intermittent oil burner operates at its maximum capacity as to running time only on the coldest days of the year, and on milder days releases heat at

TABLE 4. DIVISION OF DETERMINATIONS OF FLUE GAS TEMPERATURES AND CO_2 AS TO AGE OF INSTALLATION

	Steam	Hot Water	Warm Air	Weighted Av.
3 Yrs. or Less				
Flue Gas temp.	650	620	615	635
CO_2	7.5	7.7	7.9	7.7
No. Installations	45	14	24	83
Older Than 3 Yrs.				
Flue Gas temp.	635	600	680	633
CO_2	8.2	8.2	7.5	8.0
No. Installations	32	15	9	56

TABLE 5. DRAFT READINGS OBTAINED IN THE SMOKE PIPE AND IN THE COMBUSTION SPACE

Draft Inches of Water	Steam		Hot Water		Warm Air		Total	
	Front	Rear	Front	Rear	Front	Rear	Front	Rear
Below 0.02	6	3	1	1	–	–	7	4
0.02 - 0.05	26	10	7	5	5	6	38	21
0.05 - 0.10	11	27	2	14	5	18	18	59
0.10 - 0.15	4	19	1	4	2	8	7	31
Above 0.15	1	15	–	1	–	2	1	18
	.048	.111	.04	.07	.06	.086	.048	.0965
	48	74	11	25	12	34	71	133

the same rate but for fewer total hours. In hand firing the maximum rate is only used on the coldest days, consequently a lack of heating surface is only extremely serious at those times. In oil burning, it is serious at all times as the burner is releasing heat at the maximum rate. The lack of adequate heating surface is further aggravated by the tendency to adjust the burning rate in excess of the maximum heat requirements. Some of the reasons for this are: lack of knowledge of maximum heat requirements, necessity for pick-up capacity for lowered night temperatures, fitting the burning rate and brick work to the available combustion space rather than to the heating load, and the

greater ease with which complete combustion can be obtained at a high burning rate than at a low rate.

As an example of the overloading of a boiler the case of a larger installation than any included in the present survey may be cited. The boiler is an 8-section square boiler originally used with hand firing of coal. When converted to oil the entire combustion space was used for oil burning which allowed the burning of about 13 gal per hour based on 3 cu ft/gal. The flue gas temperature obtained on test was 920 F after a few minutes of operation and the CO_2 was 8.5 per cent, which indicates a 36 per cent stack loss during running periods. The boiler required an additional 5 sections to burn efficiently 7 gal per hour, which would be ample for the heat requirements. The flue gas temperature under these conditions would probably be 450 F and the heating surface would be about 35 sq ft per gal of fuel burned per hour.

The flue gas temperatures shown in Table 3 were taken at the boiler or furnace outlet or connection to the smoke pipe. Extra heating surface com-

TABLE 6. TABULATION OF METHODS OF SUPPLYING DOMESTIC HOT WATER

Types	Steam	Hot Water	Warm Air	Total
Fire Coil	8	9	18	35
Gas	14	10	13	37
Coal	2	0	0	2
Indirect	36	4	0	40
Oil	6	3	1	10
None	8	2	2	12
Total	74	28	34	136

monly termed heat savers was installed in one steam system, in two hot water systems, and three warm air systems. The average drop in temperature through the heat savers was 258 F. For the three warm air systems, this drop was 365 F. The average flue gas temperature for all installations was 625 F when the temperatures taken after the heat savers in these six cases are used, as compared to an average of 634 F when the temperatures at the outlets of the boilers and furnaces are used in all cases. The values on Fig. 8 are those obtained at the boiler or furnace smoke connection. In one case with added heat absorption surface the loss to the chimney was 10 per cent, one per cent lower than the loss quoted in connection with Fig. 8.

Table 4 shows a summary of the survey results as to CO_2 and flue gas temperature separated as to installations made in the three years before the survey and those made previous to that time. For the newer installations the flue gas temperature averaged 635 F and the CO_2 7.7 per cent. These values for the older installations were 633 F and 8.0 per cent. The separation was based on the time of burner installation. In some cases this coincides with the time of the heating plant installation and in other cases the burner was placed in an older heating plant. The results indicate no great change in the efficiency of the newer installations.

The averages of 7 boiler-burner units recently installed were found to be 529 F flue gas temperature and 9.3 per cent CO_2. Two installations of one manufacturer showed an average flue gas temperature of 445 F and 7.2 CO_2, and four of another manufacturer showed an average of 629 F and 11.3 per cent CO_2. In each case this corresponds to about a 21 per cent stack loss.

The draft readings obtained in the combustion space and at the smoke pipe connection are shown in Table 5. The draft loss through 12 warm air plants where the draft was observed at the two locations averaged 0.026 in. H_2O. In 43 steam plants the draft loss was 0.042 in. H_2O and in 8 hot water plants was found to be 0.037 in. H_2O. Usually a high reading in the combustion space would be associated with a low CO_2 reading. The particular weather condi-

TABLE 7. TABULATION OF FUEL COSTS ON A ROOM AND SQUARE FOOT OF FLOOR SPACE BASIS

	Steam		Hot Water		Warm Air		All Systems	
	Oil	Solid fuel	Oil	Solid fuel	Oil	Solid fuel	Oil	Solid fuel
No. of Systems with Comparative Cost Data	24	24	11	11	17	17	52	52
Ave. Cost per Sq. Ft. of Floor - Cents per Yr.	8.26	8.13	7.56	7.37	7.11	8.81	7.74	8.18
No. of Systems with Cost Data	55	28	18	11	26	23	100	62
Ave. Cost per Sq. Ft. of Floor - Cents per yr.	8.35	8.23	7.23	7.37	7.07	8.82	7.82	8.30
Minimum Cost Cents per Sq. Ft. per yr.	4.24	3.5	4.5	4.5	4.1	3.1	4.1	3.1
Maximum Cost Cents per Sq. Ft. per yr.	15.2	13.8	10.70	10.4	11.8	14.5	15.2	14.5
No. of Systems with Comparative Cost Data	24	24	10	10	18	18	52	52
Ave. Cost Dollars per Room per yr.	16.80	16.21	16.03	15.96	14.69	17.42	15.93	16.58
No. of Systems with Cost Data	50	28	20	10	27	21	97	59
Ave. Cost Dollars per Room per yr.	16.75	16.48	15.11	15.96	14.38	17.17	16.78	15.63
Minimum Cost Dollars per Room per yr.	8.60	10.70	7.00	10.50	11.20	9.00	7.00	9.00
Maximum Cost Dollars per Room per yr.	33.90	32.00	22.50	26.50	20.00	30.00	33.90	32.00

tions at the time of taking the reading affect the draft especially if a draft regulator is not installed. Eighty-five installations had draft regulators and 56 were without these useful devices.

The number of installations having some means of humidification was 63 as against 70 without any such means. In warm air plants, two out of 33 had no humidification method, in hot water plants 13 out of 23, and in steam plants 55 out of 77 had no means of supplying moisture to the air. Table 6 shows the means of supplying hot water for tap use. Seventeen boiler installations used coils in the combustion space as compared to 40 using indirect heaters with boiler water as the heating medium.

Yearly costs of heating were obtained in all cases where the owners had kept careful records and in many cases these were checked with fuel companies' records of deliveries. Twenty-seven reported savings which averaged 29 per cent with the changing from hand-fired solid fuel to oil burning. Sixteen reported the same cost with the two methods of heating and 16 reported an

increased cost of 23 per cent with the change from solid fuel to oil burning. In some of the cases where savings were obtained a new boiler, furnace or heat saver was installed at the time of change.

Table 7 presents the cost data on a basis of each square foot of floor space in occupied rooms and also on a basis of cost per room. Since types of construction vary greatly, the cost on either of these bases would vary considerably, even should the efficiency of the heating plant be the same in every instance. The data are shown in two ways: for only the installations in which records were obtained for the two types of fuel and for all cases where cost records were obtained. In many instances the original installation was arranged for oil burning and therefore no comparison was available.

Based on 52 installations the cost per square foot of floor space per year

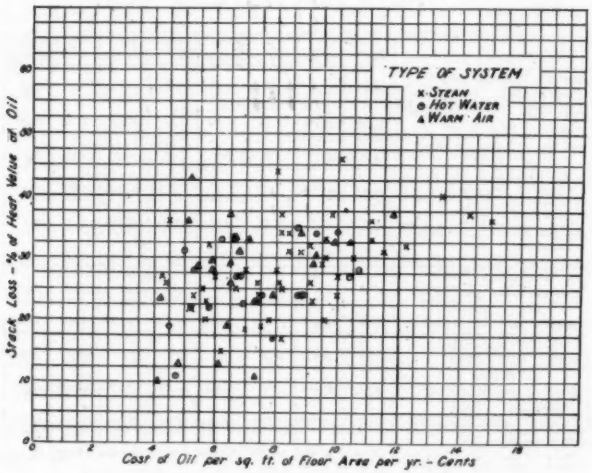

FIG. 9. RELATION BETWEEN THE COST OF OIL FOR HEATING AND THE MEASURED STACK LOSS

with oil heating was 7.7 cents and with solid fuel 8.2 cents. The cost per room per year with oil burning figured out at $15.93 and with coal $16.58 for cases reporting costs for both fuels. These costs of course depend upon the unit price of fuel during the years involved. The oil burning period of years was immediately previous to the time of the survey and the solid fuel burning previous to this variable number of years. It is, therefore, very difficult to know the unit prices of the fuels involved.[2] The figures when placed on the room basis are subject to the variations of room sizes in various types of houses. This accounts for some of the variation in costs per room, which varied from $7.00

[2] The average prices per ton for solid fuels in Madison for the 4 years 1929 to 1933 were as follows: Illinois bituminous, $8.44, semi-bituminous egg size, $11.94, coke, $12.25, anthracite pea and nut, $15.94; the arithmetical average of these is $12.14. The average prices per gallon for domestic grades of fuel oil in Madison for the 4 years 1930 to 1934 were as follows: No. 1—7.3c, No. 2—6.5c, No. 3—6.4c, and No. 4—6.1c; the arithmetical average of these is 6.6c.

to $33.90 with oil burning and from $9.00 to $32.00 with hand-fired solid fuel. The average cost of oil heating was $16.78 for the 97 cases in which oil burning costs were obtained on a room basis.

In 59 cases where solid fuel costs were given, the average cost per room was $16.63. Since the same installations are not involved in the two sides of this comparison, the size of the rooms involved would influence the results. On a square foot of floor area basis, the costs were 7.8 cents per year for oil in 100 cases as compared to 8.3 cents per year for 62 cases with solid fuel, which check very closely with the figures also given in Table 7 for the 52 cases in which oil and solid fuel costs were given for the same installations.

On Fig. 9 are plotted the points representing the cost of heating with oil and the measured stack loss. These measured stack losses were obtained from Fig. 8 by the use of the CO_2 and flue gas temperature readings for each installation in which oil heating costs were known. There would be a minimum cost of heating for every house construction even with complete utilization of the entire heating value of the oil. With a given construction the greater the stack loss the higher the cost of heating. In general the distribution of points indicates that usually a high stack loss means a high cost of heating. That a few points indicate low costs of heating in spite of a high stack loss is understandable by several reasons. The efficiency of oil burning might have been poorer at the time of test than over the period of years in which the yearly oil costs were taken. The economy of heating might be secured in certain cases by underheating parts of the house. The house may have been well insulated so that the heat losses were low or the losses may have been largely recovered as useful heat from the smoke-pipe and chimney.

CONCLUSIONS OR SUMMARY

A study of the losses during off periods of oil burner operation on a laboratory set-up indicated that a saving of 9 per cent resulted from the prevention of air circulation through the combustion space during these periods. This saving was found at about one-third load with the oil burner adjusted to run continuously when carrying full rating.

A similar study in a residence indicated under fairly average heating conditions for the locality an 8.9 per cent saving with an automatic off-period damper with a continuously maintained room temperature of 70 F, although unequal sun intensities probably make this figure slightly high. With lowered night temperatures, the saving found due to the off-period damper was 7.2 per cent. This is considered low because the damper was controlled by the room thermostat and remained open during the off periods of the warming up period. It is believed that the saving has some direct relation to the number of starts and accordingly the saving in connection with dual temperature operation would be less than with continuous room temperatures.

The off-period loss and therefore the saving possible with the use of an off-period damper depends on many variables, some of which are: the heat capacity of the brick work, the length and frequency of running periods, adjustment of the draft as to intensity, and the resistance to gravity air flow of the air supply path and of the heat absorption passes. The savings reported in this paper although applying strictly to the installations tested are considered to be an

indication of the savings possible in somewhat better than average installations. Where conditions are less favorable to efficiency the use of the damper would result in a greater saving. The question of safety arising in the need for ventilation of the combustion space during off periods to clear it of combustible gases has not been studied. This question requires considerable attention before off-period dampers might be used generally in residence oil burning.

A survey of some 140 residence installations indicated a stack loss of 27 per cent as determined by a CO_2 average of 7.8 per cent and an average flue gas temperature of 634 F. The presence of unburned fuel items in the flue gases was not determined and if present, would increase the above loss. The recovery of heat loss by conduction or radiation from the heating plant, such as from the boiler, smoke-pipe or chimney, would increase the efficiency above that indicated by the above loss.

A wide variation in stack losses was found indicating that many installations may be rated as excellent and others as unacceptably poor. The largest loss found in the survey was 46 per cent and the smallest was 10 to 11 per cent. A separation into installations made during the three years previous to the survey and those made previous to that time indicates no improvement in the average efficiency.

An analysis of costs of oil heating on a unit floor area basis indicates a relation between cost of fuel and efficiency as shown by CO_2 and flue gas temperatures. A reasonable although arbitrary standard for an efficient installation would appear to be 500 F and 10 per cent CO_2 without smoking, which corresponds to a stack loss of 18.5 per cent.

DISCUSSION

F. B. ROWLEY: The author of this paper is to be congratulated on presenting some conclusive test data which if put to practical use will save the ordinary home owner a good percentage of his heating bill. Due to the fact that it is difficult to get an oil burning nozzle that will burn small quantities of oil successfully, many small heating plants are equipped with oil burners which burn the fuel at a high rate for very short periods of time. The test data given in this paper show the inefficiencies of this method, particularly when there is no shut-off damper to prevent the heat from passing up the stack on the off period. Most owners do not realize the heat loss due to the free passage of cold air through a hot boiler. The close check between the results obtained in the field and in the laboratory indicate that every burner is a source of loss if some means is not used to shut off the draft during the off period.

As the author states, an increase in efficiency might be obtained by better adjustment of combustion. The average percentages of CO_2 shown agree very closely with the results obtained by a similar survey made at Minnesota, and no doubt represent average oil burner operation. There has been a noticeable trend in recent years for oil burner companies to give more attention to the proper installation of the burners and it is probable when the general public begins to understand what some of the heat losses are that test data such as are shown in this paper will exert an influence over the type of installation which will be accepted.

M. B. WATSON: Professor Nelson is to be congratulated on the very definite results and the clear exposition of the same which he has presented, showing the value in concrete terms of preventing the free circulation of air through oil burning

furnaces during off periods. An opportunity is given to oil burner manufacturers to obtain efficient installations by including in their standard equipment devices for this purpose.

It is unfortunate that flue gas tests were not extended to include unconsumed hydrocarbons and carbon monoxide. Apart from accounting for part of the 16.5 per cent discrepancy between theoretical and actual efficiencies, it would be extremely valuable to know to what extent combustion may be expected to be complete under different conditions of furnace temperatures and volume per gallon per hour, and what percentage of CO_2 accompanied the best compromise between losses due to excess air and those due to incomplete combustion. Also to what extent these conditions varied between cold and hot furnace. Many oil burner installers use the CO_2 indication only in the adjustment of their burners, and they take no cognizance of the furnace temperature, which may account for the wide range of efficiencies found in burners supposed to be properly adjusted.

An interesting point is brought out, of which many engineers are subconsciously aware, but to which insufficient weight is frequently given in discussing efficiencies, i.e.: "It is interesting to note that provided the outlet temperature is low the loss of efficiency due to low CO_2 is not serious. Sufficient heat absorption surface for the burner capacity used seems most essential." This feature is illustrated in Fig. 8 and indicates the fallacy of depending only on gas analysis as an indication of efficiency.

Another item of extreme importance is presented in showing the value of adequate heating surface for oil burning as compared with solid fuel boilers. In the latter case high combustion rates are obtained only occasionally throughout the year and low efficiencies at these times are immaterial, whereas oil burners operate at maximum rate in all weathers and the surface must be large enough to absorb the generated heat with high efficiency. Many boiler manufacturers recommend the selection of boilers on the basis of 10 to 25 per cent greater capacity for oil burning in the same boiler as compared with solid fuel. Based on efficiency this procedure should be reversed.

Was any relation discovered in the field survey between volume of combustion space per gallon per hour and efficiencies for different furnace linings, with a view to determining an optimum value for water-cooled-iron and for brick furnaces?

Was a similar relation discovered between heating surface and gallons per hour? Of course this relation would be of little use to engineers until manufacturers publish data on heating surfaces of their boilers.

Some very interesting deductions can be drawn from the results presented in Tables 2, 3 and 7, relative to efficiencies of steam, hot-water and warm-air systems. Table 3 shows a lower average gas temperature (probably accompanied by higher efficiency) for hot-water systems. This is to be expected due to the lower average temperature of the boiler surfaces. This lower average surface temperature might, however, be expected to cause less complete combustion; but this is not borne out by the data in Table 2 which (without considering unburned gases) indicate a higher efficiency for the hot-water systems. Also the average values in Table 7 are overwhelmingly in favor of hot-water heating, showing consistently lower average costs and much smaller limits between extremes than either warm-air or steam systems.

W. A. DANIELSON: The use of automatic draft control in the smoke pipe results in considerable loss. All the air that passes through this draft control is from the basement, and this air must eventually come from the outside. This air is heated to the basement temperature and consequently the heat used is wasted. If this draft was connected to the outside, so that only cold outside air was used in cutting down the chimney draft, the loss would be eliminated, but of course, the outside connection must necessarily be made in such a way that the wind would not cause difficulty. The CO_2 and flue gas temperature should really be taken on the chimney side of

this draft rather than in the usual way. This fact became evident to me when tests on a vented gas steam radiator were conducted. The only practicable measurement of output in a gas steam radiator is to determine the chimney losses, and this could only be done by taking the temperature and CO_2 on the chimney side of the vent.

Engineers could well work on a type of damper that would regulate the chimney draft by means other than to cool off the chimney with basement air. Another direct result of this method is that basements are ordinarily much colder than they otherwise would be.

D. W. NELSON: The 16.5 per cent difference between theoretical and actual efficiency mentioned by Mr. Watson was found on tests with a boiler in the laboratory. It is unfortunate that the analysis did not include unconsumed hydrocarbons so that this difference could be broken up into parts. A drop section in this boiler interfered with the flow of gases and probably exerted a chilling effect before the combustion of gases was complete. The other reason for this large difference of 16.5 per cent was due to the brick combustion space being built largely below the water-cooled surface and without adequate insulation. This loss is recognized in some of the later boiler designs by water-cooling the floor of the combustion space.

Mr. Watson brings to mind many items that would be well worth while determining in a laboratory study but which are beyond the practical limits of the field survey reported in this paper. Only by a very careful and extensive laboratory study could such items as the relation between the optimum CO_2 and the furnace temperature or the relation between the amount of refractory-lined surface and the efficiency be determined. It is the hope that some of these points may be investigated in later studies.

An additional reason for the better efficiency and lower cost found for hot water heating over steam heating may be due to such systems being found usually in the more substantially built homes. It has been true in the past, I believe, that hot-water systems often appear in homes built by the owner. He has been willing to put in a larger boiler than the minimum needed to carry the load. Whereas, in Table 7, in the steam and also in the warm air columns, are the costs of many plants in houses built to sell where every effort was made to reduce first costs. It is entirely possible to have any one of the three systems and have a most efficient plant. The purpose of this survey was to call attention to the relation between the cost of heating and the efficiency of the heating plant as indicated by the simple tests of CO_2 and flue gas temperature determinations.

No. 1023

THE DUST PROBLEM IN AIR CONDITIONING

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OF the various air conditions required for comfort and health, one of the most difficult to deal with in a scientific way has been that of impurities. This applies particularly to the solid material which floats in the air in small sized particles, commonly known as dust. Dust is composed of mineral, vegetable, and animal matter, and comes from practically every activity which may cause the abrasion of materials such as earth, sand, ash, rubber, stone, wood, paper, etc. The particles find their way into the atmosphere through such channels as industrial processes, household work, smoke from chimneys, moving vehicles, or any other action which may agitate the air. Dust particles will remain in suspension for periods of time, depending upon their size, density, shape, and movement of the air. Some of the finer dust may remain in the air for years.

In order to deal intelligently with dust, it is best to consider some of its properties which may make it objectionable and which may possibly be used to indicate its presence or to remove it from the air. Some of the properties which might be considered are density, size and shape of particles, color, and chemical composition. For the purpose of this discussion, the size and number of particles in a given volume of air are the most important. However, other properties, such as the tendency for small particles to aggregate into larger masses, will be considered. The effect on the human system is partly governed by the composition of the dust, and partly by the size of the particles.

Solid particles in the air are commonly classified, according to their diameters, as dust, fumes, and smoke. There is really no sharp line of demarcation between these classes. In general, dust ranges from the largest size particles which rapidly settle out of the air down to particles 1 micron in diameter. Fumes range from 1 to 0.1 micron, and smoke from 0.3 to 0.001 micron. (1 micron = 0.0004 in.) It is estimated that particles above 10 microns in diameter do not enter the human lungs and that particles below 0.5 micron in diameter will not be retained in them. It is therefore evident that, insofar as damage to the lungs is concerned, only particles which are less than 10 microns in diameter are of interest. It is generally accepted that the most dangerous sizes are from 0.5 micron to 6 microns. Larger particles of dust, such as pollen, which range

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from 14 to 60 microns, may lodge in the nasal passages, causing an irritation known as hayfever or asthma, but they do not enter the lungs. While the composition of dust has an important relation to its effect on the human system, the size of the particle determines what part of the system it may reach. The size of the dust particle is also a very important factor in its removal from the air and provides a means of measurement. Large dust particles can readily be observed floating around in the air, especially in a darkened room with a beam of sunlight passing through. It is, however, only those particles 10 microns or more in diameter that are visible to the unaided eye, and the great masses of them are invisible except with the aid of a microscope.

THE DUST PROBLEM IN VENTILATION

Some of the reasons for dust presenting a difficult phase of the ventilating problem are:

(1) It is so common, practically no air being free from it.

(2) The great majority of the particles are invisible and their presence is not apparent by a casual observation.

(3) No satisfactory method of measuring the dust in the air has been standardized and accepted.

(4) The performance to be expected of different types of cleaning apparatus is uncertain, due largely to the lack of standardized methods of measurement.

The last two factors will be specifically considered in this discussion.

Many measurements have been made to determine the number of dust particles in the atmosphere, both in various types of buildings and out-of-doors. These counts range anywhere from a few hundred up to several hundred million particles per cubic foot. After the air has been washed by a snow or rain, the count may be a matter of a few hundred, while for a reasonably dry day on a busy street it may run from four to six million particles per cubic foot. A factory building, depending upon the kind of work done, may show a count from a few thousand up to hundreds of millions of particles. Thus, the number of dust particles may vary within wide limits, and it is difficult to set standards as to the amount which should be tolerated. It is evident, however, that in most cases standards should include some consideration of the size of dust particles. A relatively small number of coarse, heavy particles might give the appearance of very dirty air and do a considerable amount of physical damage without being a menace to human health. On the other hand, large quantities of fine particles might be present which might be exceedingly dangerous to health and yet not be visible or cause any particular physical damage.

METHODS OF MEASURING DUST IN AIR

A standardized method of measuring dust is an important factor which as yet has not been settled. Several methods have been proposed and developed, although none has shown sufficient merit to warrant its being universally adopted. One common method has been to separate dust from the air and find its total weight. This has the advantage that weight is a positive measure and one that can be easily understood in making comparisons. It has the disadvantage that weight alone does not indicate the quality of the dust nor the size of the particles, and, therefore, shows nothing as to whether or not it is a

harmful dust. Total weight has no relation to the number of particles of dust contained in the air. One particle 100 microns in diameter would be equivalent to 1,000,000 particles of 1 micron in diameter. Furthermore, even though the total number of particles of dust in the air may be very high, the total weight in a given volume of air is usually very low and it is a difficult procedure to thoroughly clean a sufficient quantity of air to make a weight determination practical.

A general method which seems to give the greatest promise from the standpoint of practical application is to separate the dust particles from a given quantity of air and make an actual count of them under the microscope. In addition to giving the number of particles present, some information may be obtained as to the relative sizes and shapes of the particles. There are several possible methods of separating the dust particles from the air, but the two which seem most practical and which have been used in this investigation are:

(1) To wash the dust out with water.
(2) To collect it on a viscous coated glass plate.

An apparatus for washing the dust from the air which seems to give a very high efficiency and which is used by the United States Bureau of Public Health is the Smith-Greenberg impinger. It is designed to impinge at high velocity a stream of the dust-laden air against a plate under water. For practical purposes, this instrument has been considered to give 100 per cent collection. While the instrument has the advantage of collecting a high percentage of the dust, it has some disadvantages:

(1) It takes a considerable period of time to make a determination.

(2) Some kinds of dusts are soluble in water and, therefore, not adapted to this method of collection.

(3) In many instances, dust particles float in the air in agglomerated masses rather than in individual, fine particles. In the air, many of these particles massed together might be the equivalent of one large particle, whereas when they are passed through the water impinger, they are broken up into many fine particles. The dust count thus shows a much larger number of fine particles than actually exists in the air. The extent of this error will depend upon the nature of the dust.

After the sample is collected, the particles contained in a definite volume of the collecting water are counted under a microscope.

The viscous coated impinger principle is used in several instruments, a very common one being that designed by Dr. E. V. Hill. In this instrument, a high velocity jet of air is impinged against a viscous coated glass by the rapid stroke of an air pump. The particles are then counted under a microscope, as in the previous case. The advantages of this system are that the sample is quickly taken and the apparatus is easily handled for practical measurements. As in the Smith-Greenberg, a microscopic view of the dust particles gives much information that cannot be determined by any other method. One of the disadvantages in most such instruments is the inefficiency of the collection and, also, the difficulty of getting practical samples from concentrated dust mixtures. Comparative tests between the two types of instruments have shown that for low concentration of dust the viscous impinger will give a count equal to or greater than the water impinger, but for high concentration the water impinger will show a very much higher count. In order to make the viscous collector

applicable to practical conditions, the dust count obtained must be increased by some factor. This might not be a serious objection were it not for the fact that there can be no assurance that the factor is the same for all kinds and concentrations of dust.

IMPROVEMENTS IN THE VISCOUS COUNTER

Since there were some practical advantages in collecting the sample on a viscous coated slide as compared with collecting it in water, and since some kinds of dust apparently were not adapted to the water method of collection, it seemed worthwhile to consider the possibility of getting a higher percentage of dust collection by the viscous method. The Hill dust counter was selected as a basis for the first experimental work. In this work, three general changes were considered for improving the percentage of dust collected:

(1) To select the adhesive material for coating the glass collecting plate which would collect the largest percentage of dust and retain the dust in the best condition for counting under the microscope.

(2) To change the size and shape of the orifice and vary its distance from the viscous plate to get the most effective design for collecting a high percentage of dust which would fall within the field of the microscope for counting.

(3) To find the most effective velocity of air through the orifice and to devise a means for obtaining a constant uniform velocity of air for the sample.

The demands of an adhesive material for a suitable coating of the glass collecting plate are rather exacting and the selection has proved to be a very difficult part of the problem. The material must spread over the surface of the glass with a smooth, even surface. It must be viscous enough to catch and retain the fine dust particles and be sufficiently stable to retain them in position after the sample is taken, and yet not be so rigid that the particles will not adhere to the surface.

In trying out various materials, it was found that such materials as vaseline had an advantage in that they would maintain their position on the slide and not flow after the sample was taken, but they had disadvantages in that it was difficult to get a good smooth surface, and that the percentage of dust retained on the surface was very small. With many of the oils, the film would break and tend to form little drops on the surface of the plate. In some cases, this break would occur immediately, and in other cases it would break during the taking or counting of the sample.

A list of the oils tried out, together with some of the difficulties met with, follows:

(1) *Vaseline*—Uneven surface, very poor retention of dust.
(2) *Paraffin Oil*—Surface had tendency to break during counting.
(3) *Paraffin Oil mixed with Xylene*—Some improvement over paraffin oil, but surface tended to break.

The following oils broke either immediately or while taking the sample: (4) *Glycerin;* (5) *Olive Oil;* (6) *Castor Oil;* (7) *Cod Liver Oil;* (8) *Cottonseed Oil;* (9) *Raw Linseed Oil;* (10) *Boiled Linseed Oil;* (11) *Havoline Motor Oil;* (12) *Red Atlantic Lubricating Oil;* (13) *Lard Oil;* (14) *Marvel Penetrating Oil;* (15) *Raw Paraffin Oil.*

(16) *Canadian Balsa*—Surface hardens and does not catch dust.
(17) *Sendac Floor Oil*—Difficult to focus particles but has possibilities.
(18) *Oil of Cedar*—Evaporates, leaving residue of particles which are mistaken for dust.
(19) *Filter Oil*—Reasonably satisfactory.

After many trials, the Filter Oil, No. 19 in the list, was selected as the most satisfactory. While this oil does not give a perfect coating, it was used throughout the remainder of this experimental work.

In changing the orifice, the object was to select the combination of orifice design and position of orifice relative to the plate which would give the most efficient dust collection, and a sample of such size and shape that all or a definite percentage could be brought within the field of the microscope for counting at the desired magnification.

Various kinds of orifices were used, including thin plate, short tube, rounded entrance, and narrow slits each of different areas and at different distances from the plate. A cylindrical orifice 0.04 in. in diameter through a ⅛ in. plate with 1/16 in. clearance was finally selected.

For the first changes in the orifice and the coating of the viscous plate, the regular 5 cu in. suction pump was used to draw the air through the orifice against the viscous coated glass slide. The progressive changes improved the percentage of dust collection to such an extent that it was difficult to count the dust particles obtained without so reducing the stroke of the pump as to make the apparatus impractical for average dust concentration. Pumps with smaller cylinder diameters were substituted, but still many samples were overloaded with dust particles, and there was the further difficulty that the velocity of the air impinging against the surface of the slides varied from zero to a maximum with an uncertain average.

From experimental work previously reported,[1] it was definitely known that the air velocity greatly affected the percentage of dust collection. The hand pump was therefore discarded and the air was drawn at constant velocity through the orifice by means of a small vacuum pump. The rate of air flow giving the best collection of dust was determined by trial. While this method was an improvement, it was still difficult to get samples of air which were sufficiently small to make the counting practical. The dust samples were concentrated in the central part of the field while the outer portions were practically clear. There appeared to be at least two ways out of this difficulty:

(1) To devise some type of shutter which would operate accurately on the split second to admit very small samples of air.

(2) To devise an arrangement for moving the orifice relatively to the viscous coated plate at a predetermined speed to distribute the dust sample over the field.

Both methods were tried but the second was selected as the most promising, and the counter was developed on this principle.

In its present form, the counter is designed so that the air is drawn through the orifice at a constant velocity by means of a vacuum to impinge on a viscous coated glass plate. The amount of air striking the plate at any given section is regulated partly by the size of the orifice and the air velocity through the orifice, but finally by moving the orifice parallel to the surface of the collecting plate at a predetermined constant velocity. By this arrangement, a continuous uniform path of dust particles is deposited on the plate. The width of the path is within the field of the microscope and by counting the particles for a given length of the dust path, the number per given air volume is determined.

[1] Determining The Quantity of Dust in Air by Impingement, F. B. Rowley and John Beal, A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 483.

The instrument in its present form is provided with an 0.04 in. diameter orifice through a ⅛ in. thick plate. The plate is placed 1/16 in. from the viscous coated glass slide. For the best results, the air is drawn through the orifice by a vacuum equal to 5 in. of mercury, and for average laboratory air a countable

TABLE 1. RESULTS OF DUST COUNTS TAKEN AT DIFFERENT SECTIONS OF PATH

SAMPLE NO.	DUST COUNT SECTION A	SECTION B	SECTION C	AVERAGE	PARTICLES PER CU FT
1	230	210	198	212	1,293,000
2	293	326	300	306	1,865,000

The complete counts for the three sections of Sample No. 2 are given:

SECTION A							
7	4	9	10	4	1	2	
4	6	7	9	7	1	2	
5	2	9	10	4	5	7	
2	1	6	10	12	5	4	
3	6	8	12	10	4	9	
2	4	8	9	7	7	6	
3	6	10	10	5	7	2	TOTAL: 293
SECTION B							
2	10	3	14	4	5	3	
5	7	8	4	12	8	0	
4	9	8	13	13	10	1	
0	8	10	9	17	2	5	
5	9	7	12	11	1	1	
3	8	6	9	8	7	2	
2	7	8	9	10	2	5	TOTAL: 326
SECTION C							
0	2	4	6	11	3	2	
4	4	4	6	15	4	4	
1	10	5	17	6	8	0	
0	3	6	9	9	5	17	
8	2	7	9	9	6	8	
2	4	4	9	15	9	1	
3	5	4	6	11	5	8	TOTAL: 300

sample can be obtained by moving the orifice over the surface of the glass at the rate of ¾ to ¼ in. per second. This gives a volume of air of approximately 0.6 cu in. per millimeter of length for the dust path and a velocity through the orifice of 375 fps. Counts are taken for different sections of the path to give a fair average for the sample. Table 1 shows the results of counts taken at different sections of path for several samples.

The advantages of this method of sampling the air are:

(1) That the air is made to impinge against surface of the collecting plate at a uniform velocity throughout the sample.

(2) The sample from a small but definite quantity of air is spread in a uniform path on the surface of the glass and falls completely in the field of the microscope.

With an orifice which remains stationary relative to the collecting surface, it is impossible to get uniform conditions of air flow for the sample when dealing with air volumes of less than 1 cu in. Furthermore, when the sample is collected, it is concentrated in the center of the field, making it impossible to count the dust in the center squares accurately, while there are no particles at all in the outer squares of the field.

Table 2 shows the counts from two average samples of air, one taken with

TABLE 2. COMPARATIVE COUNTS FOR SAMPLES TAKEN WITH A STATIONARY AND A MOVING ORIFICE

STATIONARY ORIFICE							TOTAL	
0	0	2	4	4	0	0	10	
1	2	5	8	5	6	0	27	
1	0	14	20	10	4	3	52	
3	5	12	18	12	7	4	61	
3	8	8	11	10	5	5	50	
2	1	4	1	4	3	2	17	
0	3	5	1	3	1	0	13	230
MOVING ORIFICE								
0	2	5	10	8	5	1	31	
0	6	10	9	3	3	1	32	
2	4	10	4	5	1	1	27	
0	6	4	7	10	6	1	34	
1	3	10	10	10	7	2	43	
0	2	16	10	8	3	2	41	
1	2	10	11	6	2	0	32	240

a stationary orifice and the other with a moving orifice. From this it will be observed that for the stationary orifice 119 or 51.7 per cent of the particles are collected in the 9 center squares, leaving only 48.1 per cent for the remaining 40 squares, while for the moving orifice the distribution is made more uniformly.

COMPARATIVE DUST COUNTS WITH TWO TYPES OF IMPINGERS

Accurate comparisons between dust counts made with the water impinger and the viscous impinger developed in this work are difficult to make, due to such variables as:

(1) The possible shattering or breaking of agglomeration of dust particles by the water impinger, thus giving more particles than are actually in the air.

(2) The possibility that some of the dust particles may be soluble in water.

(3) The difference in counts obtained by letting the dust settle in the counting cell for different lengths of time.

(4) There is some question as to whether the very fine particles may be counted as easily when in water as when on the dry viscous plate.

During the first comparative tests, city water was used and a control count taken for the water impinger. By comparing these results with those obtained when using distilled water, the results were found to be very erratic and sometimes the count was two to three times higher when using city water. This variation was entirely too much to be accounted for by the initial dust in the water and seemed to be due to the precipitation of solids during the test. Since the logical procedure was to use distilled water, the main value of the comparison between the distilled and city water was to show the precautions which

TABLE 3. COMPARISONS BETWEEN DUST COUNTS FOR SAMPLES OF THE SAME AIR BY THE WATER IMPINGER AND THE VISCOUS IMPINGER AT 100 DIAMETERS MAGNIFICATION

Test No.	Date	Particles per Cu Ft		Ratio of Count Viscous Water	Notes
		Viscous Impinger	Water Impinger		
1	3/26/'35	1,132,000	580,000	1.95	Air from main lab.
2	3/26/'35	637,000	290,000	2.19	Air from main lab., after passing through filter.
3	3/27/'35	822,000	467,000	1.76	Air in main lab.
4	3/28/'35	1,220,000	252,000	4.84	Taken in main lab. about 10:00. a.m.; snow starting outside.
5	3/28/'35	612,000	201,500	3.04	Taken in main lab. about 2:00 p.m., after snowfall.
6	3/28/'35	377,000	166,800	2.26	Taken with outside air from 2nd floor of bldg. about 3:00 p.m.
7	3/29/'35	318,000	233,000	1.36	Taken in main lab.
8	4/15/'35	548,000	495,000	1.11	Taken with outside air from 2nd floor of bldg. Large wind-blown particles more easily detected in Smith-Greenberg, reducing ratio.
9	4/18/'35	2,010,000	412,000	4.86	Taken in main lab. Janitor began sweeping, increasing the dust concentration.
10	4/18/'35	1,431,000	106,100	13.47	Taken in 2nd floor room. Particles very small; still air conditions.

must be taken in making such counts. In all of the tests reported here, distilled water has been used.

While there was a considerable variation in the ratios between the counts obtained by the two methods, this ratio was always in favor of the viscous impinger and usually ranged 2:1 or better. Some of the comparative results obtained between the two instruments are shown in Table 3. These tests were taken when there was a considerable amount of fine dust in the air due to the dust storm in the surrounding country. It is possible that the high ratio in favor of the viscous impinger is due either to the fact that some of the fine dust is soluble in water and therefore does not show up in the count by the water impinger method, or perhaps the very fine particles are not visible when

in the water solution. It is particularly interesting to note the ratios obtained from Tests 4, 5, and 6 which were taken on March 28. On the evening of March 27, the outside air was particularly dusty, and on the morning of March 28, just prior to the time that Sample 4 was taken in the Main Laboratory, a snowstorm set in which lasted intermittently throughout the morning. Sample No. 4, which was taken in the Laboratory shortly after the snowstorm started, probably represented the outside air with the fine dust of the night before. This showed a very high ratio in favor of the viscous impinger.

Sample No. 5 which was taken about 4 hours later would naturally represent a mixture of the original Laboratory air with some of the cleaner air filtered in from the outside. This air showed lower results for both instruments, but still a rather high ratio in favor of the viscous impinger. Sample No. 6 was taken shortly after Sample No. 5, with outside air. This was much cleaner than the Laboratory air by both methods, but still it showed a fairly high ratio in favor of the viscous impinger.

If it is assumed that the air for Test No. 5 was a mixture of the Laboratory air for Test No. 4 and outside air for Test No. 6, calculations may then be made to show the percentage of Laboratory air by either test method. If $X =$ the percentage of Laboratory air in Sample No. 5, then for the viscous counter $1{,}220{,}000X + 377{,}000 \times (1.00 - X) = 612{,}000$, and $X = 28.8$ per cent of Laboratory air in Sample No. 5.

For the water impinger, $252{,}000X + 166{,}800 \times (1.00 - X) = 201{,}500$ or $X = 24.5$ per cent as the original Laboratory air in Sample No. 5. The close agreement of the percentages of air as calculated by the two methods is worth considering.

The above counts were taken by using a microscope of 100 magnifications, although similar relations were shown with 50 magnifications, as will be discussed later in the efficiency tests of filters. In every case, there was a greater quantity of dust shown by the dry viscous impinger.

Since the viscous counter takes practically an instantaneous sample there will be some variations due to the natural variation of dust count in the air. Table 4 shows the results from samples taken at successive, short intervals of time. The variation in count is not excessive but indicates that for average conditions more than one sample should be taken.

RATING THE EFFICIENCY OF DUST CLEANERS

The rating of dust cleaning apparatus has always been a troublesome problem, largely due to the fact that no standard method has ever been adopted for measuring the amount of dust in the air. For several years, a committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS worked on the problem and, after considering the various methods available for measuring dust, finally prepared the A. S. H.V. E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work in which a given kind of dust was used and the percentage of total weight retained by the filter taken as a measure of its efficiency.

The dust prescribed in this code consists of a mixture of 50 per cent by weight of powdered lamp black and 50 per cent by weight of ashes of Pocahontas bituminous coal screened to pass the 200-mesh sieve. Since the code is

for the purpose of determining the dust-holding capacity of the filter as well as its efficiency or arrestance, it must necessarily provide an accelerated test. 0.65 grams of dust per 1000 cu ft of air are therefore prescribed. The mixture of dust is fed into the air by a specially designed dust mixer, and the amount passing through the filter is obtained by thoroughly cleaning a certain fraction of the air passing through and weighing the dust thus obtained.

This code serves a useful purpose, and the following comparisons are not to criticize it but rather to examine some parts of it, together with the results

TABLE 4. VARIATION IN COUNT FOR DUST SAMPLES TAKEN WITH THE VISCOUS COUNTER AT SUCCESSIVE INTERVALS OF APPROXIMATELY 2 MIN

DATE	SAMPLE NO.	DUST PARTICLES PER CU FT	PER CENT VARIATION FROM AVERAGE
3/6/'35	1	836,000	+ 15.0
	2	680,000	− 6.5
	3	600,000	− 17.5
	4	793,000	+ 9.1
	AVERAGE:	727,000	
3/7/'35	1	1,000,000	+ 12.2
	2	829,000	− 6.9
	3	845,000	− 5.2
	AVERAGE:	891,000	
3/11/'35	1	418,000	+ 7.7
	2	290,000	− 25.2
	3	376,000	− 3.1
	4	469,000	+ 20.8
	AVERAGE:	388,000	
3/12/'35	1	327,000	− 2.9
	2	312,000	− 7.4
	3	372,000	+ 10.4
	AVERAGE:	337,000	

obtained by its use, to see whether or not it provides everything that is desirable in the rating of an air filter. There is a question as to whether the particular dust prescribed, together with the method of feeding it in concentrated form to the filter, represents practical conditions. These dust particles may have a tendency to remain in agglomerated masses and not to separate out into individual, fine particles as they would in practice. If this should be the case, certain types of filters might collect a much larger percentage of the weight of dust in the air during a performance test than they would during practical operations.

On the other hand, there may be the question of overloading certain viscous types of filters and not giving the oil-coated fibers the same chance of collecting the dust that they would have in normal operation. Next, there is the question that even if the arrestance by test represents a true percentage of the weight of

dust which would be taken out in normal operation, there is still a doubt as to whether weight alone gives everything that is desirable when considering filter efficiency. It is evident that a few large particles which would be taken out easily by a filter may represent a large percentage of the weight of a given mass of dust and yet may not at all be representative of the dust in the mixture or of the dust which is or is not taken out by the filter. If these dust particles were 10 or more microns in diameter, each one of them would equal the weight of 1000 or more of particles 1 micron in diameter. The filter might be taking out the large particles, thus showing a very creditable efficiency, while it would be passing all of the very fine particles which might be the real dangerous dust insofar as ventilation is concerned.

In order to check the possible difference in the rating of filters by the

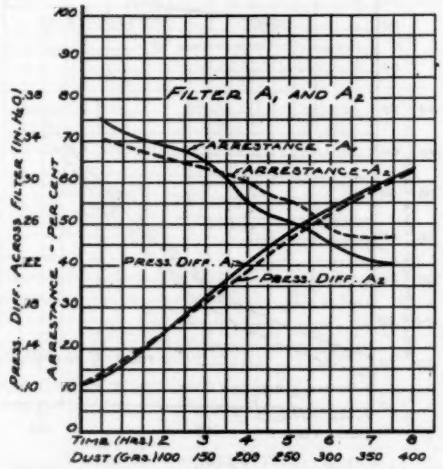

FIG. 1. TEST RESULTS OF FILTER *A*

A. S. H. V. E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work and by the dust count method, four typical filters have been selected and designated as Filters *A*, *B*, *C*, and *D*. Filters *A* and *B* were rated by the standard code method and the ratings are shown by the curves of Figs. 1 and 2. Filter *C* has been tested and found to be from 95 to 98 per cent efficient for dust particles 14 microns in diameter or greater. The 4 filters were placed in the standard test apparatus and their efficiencies determined by the dust count method when using both the water and viscous impinger. The tests were not made with the heavy dust concentration as prescribed by the code, but rather by passing the normal laboratory air through the filters and taking dust counts of the entering and leaving air. The results of these tests are shown in Tables 5, 6, 7 and 8.

By comparing the test results for Filters *A* and *B* when using the two different methods, it will be noted that for Filter *A* the standard code gave an

efficiency of 72 per cent for the filter when new, while the dust count method gave an efficiency of 41.5 per cent. For Filter *B* a test by the standard code gave an efficiency of 91 per cent for the new filter, while the dust count method gave an efficiency of 52 per cent. The ratio of the efficiencies by the dust count method to the standard code method is 57.6 per cent for Filter *A* and 57 per cent for Filter *B*. The rating curves for Filters *C* and *D* by the standard code method are not available. However, Filter *C* was found to have from 95 to 98 per cent efficiency when using lycopodium powder, the size of particles being about 14 microns.

The comparative test results for the different filters would indicate that the efficiency of the average filter in normal operation might be expected to be

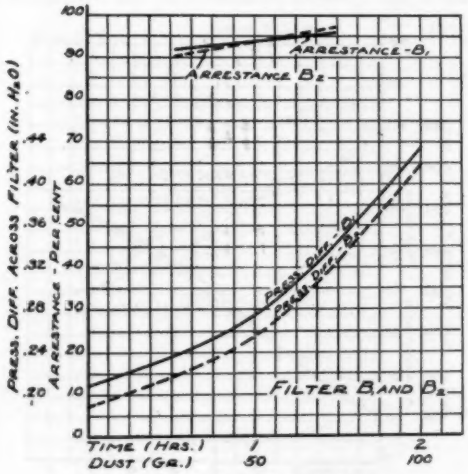

FIG. 2. TEST RESULTS OF FILTER *B*

lower when determined by the dust count method than when determined according to the standard code. It seems reasonable to believe that this difference is partly due to the fact that the filter is more efficient for large than for small particles. The number of particles counted in any case will depend upon the magnification. For a magnification of 100 diameters, particles of 1 micron are easily counted, and a study under different magnifications indicates that on the viscous coated slides particles of ½ micron in diameter may be included. This varies with the individual making the count, but as a lower limit it appears that particles ½ micron in diameter may be included. At 50 diameters magnification the same study indicated that particles 1 micron in diameter may be included. For Filter *A* the efficiencies were taken by using both 50 and 100 diameter magnifications. The final results were substantially the same.

In order to determine the differences in numbers of particles counted when using 50 and 100 diameter magnifications, several comparative counts were made for different samples without moving the sample under the microscope. The results are shown in Table 9. While there is some variation in the percentages

obtained by 50 diameters magnification, the average is 48 per cent of the number obtained by 100 diameters magnification. This would indicate that of the average dust floating in the air, approximately 50 per cent of the particles visible at 100 diameters magnification are 1 micron or below in diameter. Since substantially the same efficiencies were obtained on the filters when using 50 and 100 diameters magnifications, it appears that the filter must be taking out approximately the same percentage of the particles of the different diameters.

As a further check on the reliability of requirements as set up by the standard test code, a sample of each, the carbon black and Pocahontas Ash as used in the standard code, was investigated to determine the size of particles when

TABLE 5. EFFICIENCIES OF FILTER *A* BY DUST COUNT METHOD

TEST NO.	DATE	TEST METHOD	MAGNIFICATION	COUNT		EFF. %	NOTES
				Upstream	Downstream		
1	3/22/'35	Viscous	100	2,010,000	1,144,000	43	
2	3/23/'35	Viscous	100	853,000	659,000	22	
3	3/23/'35	Viscous	50	495,000	339,000	32	Same sample as for Test 2
4	3/24/'35	Viscous	100	1,647,000	1,165,000	29	
5	3/25/'35	Viscous	50	1,020,000	665,000	35	Same sample as for Test 4
6	3/26/'35	Viscous	100	1,133,000	636,000	44	
7	3/26/'35	Viscous	50	620,000	322,000	48	Same sample as for Test 6
8	3/26/'35	Water	100	580,000	290,000	50	Taken same time as No. 6
9	3/26/'35	Water	50	294,000	115,000	61	Same sample as for Test 7
10	3/27/'35	Viscous	100	980,000	580,000	41	
	3/27/'35	Viscous	50	790,000	448,000	43	
11	3/29/'35	Viscous	100	1,077,000	569,000	47	
12	3/30/'35	Viscous	100	514,000	245,000	52	
13	3/30/'35	Viscous	100	358,000	237,000	34	
					AVERAGE:	41.5	

collected by the two different instruments and, also, when mixed in the air for the filter tests.

For this examination, each dust was fed separately into the air stream of the test apparatus by the standard dust-mixing device, and samples were taken by both instruments and also by glass slides placed in the air stream. The purpose of using the glass slides was to collect the dust particles in the condition in which they would reach the test filter, that is, without danger of breaking up the aggregations of particles. In each case, the dust was collected on a dry glass plate, an oil coated plate, and both impingers. They were examined under the microscope to make an estimate of the percentage of the different

TABLE 6. EFFICIENCIES OF FILTER *B* BY DUST COUNT METHOD

TEST NO.	DATE	TEST METHOD	MAGNIFICATION	COUNT		EFF. %	
				Upstream	Downstream		
1	4/2/'35	Viscous	100	764,000	363,000	53	
2	4/2/'35	Viscous	100	971,000	662,000	32	
3		Viscous	100	1,140,000	484,000	57	
4		Viscous	100	1,314,000	784,000	40	
5		Viscous	100	2,735,000	1,033,000	62	
6		Viscous	100	2,095,000	653,000	69	
					AVERAGE:	52	

TABLE 7. EFFICIENCIES OF FILTER *C* BY DUST COUNT METHOD

TEST NO.	DATE	TEST METHOD	MAGNIFICATION	COUNT		EFF. %	NOTES
				Upstream	Downstream		
1	4/4/'35	Viscous	100	3,580,000	870,000	76	200 fpm
2	4/4/'35	Viscous	100	5,770,000	2,480,000	57	200 fpm
3	4/4/'35	Viscous	100	2,830,000	1,075,000	62	200 fpm
4	4/4/'35	Viscous	100	2,420,000	1,407,000	42	200 fpm
5	4/4/'35	Viscous	100	952,000	617,000	35	200 fpm
6	4/5/'35	Viscous	100	1,418,000	571,000	60	200 fpm
7	4/5/'35	Viscous	100	1,475,000	466,000	68	200 fpm
8	4/8/'35	Viscous	100	1,343,000	625,000	53	90 fpm
9	4/8/'35	Viscous	100	1,109,000	528,000	52	90 fpm
10	4/8/'35	Viscous	100	835,000	256,000	69	45 fpm
11	4/8/'35	Viscous	100	1,047,000	528,000	50	45 fpm
					AVERAGE:	56.7	

TABLE 8. EFFICIENCIES OF FILTER *D* BY DUST COUNT METHOD

TEST NO.	DATE	TEST METHOD	MAGNIFICATION	COUNT		EFF. %	NOTES
				Upstream	Downstream		
1	4/16/'35	Viscous	100	1,384,000	584,000	58	250 fpm 0.19 pres. drop
2	4/16/'35	Viscous	100	1,485,000	806,000	46	
3	4/19/'35	Viscous	100	1,992,000	1,590,000	20	
4	4/19/'35	Viscous	100	852,000	468,000	45	
5	4/19/'35	Viscous	100	900,000	645,000	28	
					AVERAGE:	39.4	

sized particles collected. The results of this investigation are shown in Tables 10 and 11.

Referring to Table 10, the samples of carbon black showed substantially the same results by the viscous and water impingers. For the Pocahontas Ash, there appeared to be a larger per cent of fine particles in the sample collected by the viscous impinger. This may have been due to a less efficient collection by the water impinger to the solubility of some of the dust, or perhaps the finer particles were invisible in the water solution. The total number of fine particles for both types of dust appears to be fairly satisfactory.

Referring to Table 11, it will be observed that the samples collected on the

TABLE 9. RATIO BETWEEN TOTAL NUMBERS OF PARTICLES COUNTED AT 50 AND AT 100 MAGNIFICATION FOR BOTH WATER AND VISCOUS IMPINGER METHOD

Test No.	Date	Method	Location	Magnification		Percentage at 50 Mag.
				100	50	
1	3/23/'35	Viscous	Upstream	854,000	328,000	38
2	3/23/'35	Viscous	Downstream	660,000	224,000	34
3	3/25/'35	Viscous	Upstream	1,648,000	1,022,000	61
4	3/25/'35	Viscous	Downstream	1,165,000	666,000	57
5	3/26/'35	Viscous	Upstream	1,205,000	384,000	32
6	3/26/'35	Viscous	Upstream	970,000	482,000	50
7	3/26/'35	Viscous	Downstream	279,000	142,000	51
8	3/26/'35	Viscous	Upstream	1,134,000	620,000	55
9	3/26/'35	Viscous	Downstream	636,000	322,000	51
10	3/26/'35	Water	Upstream	580,000	294,000	51
11	3/26/'35	Water	Downstream	290,000	115,000	40
12	3/27/'35	Viscous	Upstream	980,000	524,000	53
13	3/27/'35	Viscous	Downstream	580,000	296,000	51
					AVERAGE:	48

glass slides show the average diameter of particles to be decidedly greater than for the samples collected by either of the impingers. The oil coated slides collected a much smaller percentage of fine particles than did the dry slides. In the case of Pocahontas Ash, the oil coated slides showed 65 per cent of the particles to be 5 microns or above, while the viscous impinger showed only 4 per cent of 5 microns or above. The percentage of fine particles of carbon black collected on the glass slide was more nearly equal to those collected by the impingers, although there was still a very much larger percentage of coarse particles obtained. While the collection on the glass slides cannot be considered to be 100 per cent perfect, it does show that the dust particles are not broken up and as finely divided in the air stream as they should be. This is quite likely one reason why the filters tested by the standard code method show a higher efficiency than when tested by the dust count method under operating conditions.

From the results obtained in the investigation, it would seem reasonable to give some consideration to the standard method of testing air filters to see whether or not the code provides everything that is required of a filter in

TABLE 10. SAMPLES OF STANDARD DUST COLLECTED BY THE VISCOUS FILTER AND THE WATER IMPINGER FOR EXAMINATION UNDER THE MICROSCOPE TO DETERMINE THE PERCENTAGES OF DIFFERENT SIZED PARTICLES IN SAMPLES

	VISCOUS IMPINGER	WATER IMPINGER
CARBON BLACK		
2 Microns or less, percentage	80	85
2 to 5 Microns, percentage	15	14
5 Microns or over, percentage	5	1
POCAHONTAS ASH (THROUGH 200-MESH SIEVE)		
2 Microns or less, percentage	82	50
2 to 5 Microns, percentage	14	40
5 Microns or over, percentage	4	10

normal operation. The important factor in rating a filter is often not that it takes out a certain percentage of dust by weight, but rather that it will take out certain objectionable dusts. In some cases, it may be the very fine dust and, in other cases, it may be the coarser particles that should be removed. The efficiency of a filter rated on the weight basis might have no relation to its efficiency when rated by the dust particles as found in normal air. A count method is easily applied to a filter either under test conditions or in normal operation, and there seems to be no reason why an efficient and adaptable counting instrument cannot be designed.

The rating of filters for different sized dust particles might be effected by estimating the percentages of dust particles of different diameters under a high magnification, or the magnification might be reduced to such an extent as to

TABLE 11. ESTIMATED SIZE OF PARTICLES COLLECTED ON GLASS SLIDES PLACED IN THE AIR STREAM OF STANDARD FILTER TEST APPARATUS

POCAHONTAS ASH	DRY GLASS SLIDE	OIL COATED SLIDE
2 Microns or below, percentage	20	Negligible
2 to 5 Microns, percentage	45	35
5 to 10 Microns, percentage	25	45
Above 10 Microns, percentage	10	20
CARBON BLACK		
2 Microns or below, percentage	55	30
2 to 5 Microns, percentage	30	50
5 to 10 Microns, percentage	10	10
Above 10 Microns, percentage	5	10

make only those particles visible which are in the range of sizes required of the filter. For instance, if a filter were to be used to clean the air of fine particles of silica dust which might enter the lungs, the dust count would be made under 100 magnification. If, on the other hand, the filter were to be used to clean out only the large particles, such as might be required for hay fever patients, then the dust count might be made at 10 or 20 magnification and the finer particles eliminated from consideration. Such a procedure would greatly facilitate the testing of filters in actual operation. It would not necessarily replace the present code but would be a valuable supplement to it and would give much desirable information as to filter efficiency.

ACKNOWLEDGMENT

The funds for this investigation conducted at the Experimental Engineering Laboratories were provided by the Graduate School of the University of Minnesota, and the studies were carried on with special reference to methods of measuring dust and the rating of air cleaning devices.

DISCUSSION

H. C. MURPHY (WRITTEN): Professor Rowley's paper represents in my opinion a worthwhile addition to the available knowledge of dust counting procedures. I have been asked to comment on this paper and I find little to criticize and much to commend.

It might be worthwhile at this time to re-emphasize the fact pointed out by Professor Rowley in this paper that the Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, which was adopted by the Society last year as a standard, was not intended and is not, of course, adapted for measuring the amount of dust in the air, neither was it proposed for testing air cleaning devices used to remove the microscopic dusts which are generally considered significant in *dust diseases.* Rather as noted in the title of this code it is distinctly for use in *general ventilation work.*

In order to avoid misunderstandings with this code it was thought advisable to re-state this in the first paragraph of the code under *Scope.*

A. SCOPE

"This Code is for the laboratory investigation and rating of devices used in general ventilation work for the sole purpose of removing solid impurities from the air. It is not intended for use in the rating of cyclones, dust separators or for devices used in the field of industrial hygiene. Ratings established under this Code are not to be confused with operating efficiencies in actual services—the dust concentration, the nature of the dust, the relative humidity and many other factors have a definite bearing upon the results obtained in actual service."

In other words this code undertakes in general to provide standards for comparing the effectiveness of commercial air cleaning devices against *nuisance dusts,* that is, broadly speaking, dusts which are large enough and heavy enough to form objectionable coatings on desks, furniture, etc., and which present the principal problem in ordinary ventilation systems. Owing to the size of these dust particles they are not ordinarily considered significant in the so called *dust diseases.*

The Committee now has in preparation a code for testing and rating air cleaning devices used in dust hazardous occupations, which it hopes shortly to present to the Society for its consideration.

Another valuable feature of Professor Rowley's paper is to point out the difficulty which is experienced in any attempt to correlate results obtained by different test procedures and to further emphasize that these procedures are intended simply as a yard stick to rate the various air cleaning devices. These ratings should not be confused with operating efficiencies in actual service as these will vary with the size, the kind, and the concentration of the dust as well as many other variables.

No. 1024

CHICAGO STANDARDS FOR CERTIFIED AIR CONDITIONING

By SAMUEL R. LEWIS * (*MEMBER*)
CHICAGO, ILL.

THE purpose of this paper is to describe the Standards for Certified Air Conditioning developed for the Chicago district by representatives of The Consulting Engineers Division of the *Western Society of Engineers*, the *Chicago Master Steam Fitters Association* and the *Chicago Ventilating and Air Conditioning Contractors Association*. These standards have been adopted by the two contractors' organizations and have been endorsed in principle by the Illinois Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and they have created much interest and discussion locally as well as in other sections of the country because of their novelty and newness.

As one of the men who developed these standards I am glad to present this discussion in the hope that it will be helpful to all who are interested in the development of air conditioning along sound engineering lines.

First let me say that there is no compulsion on anyone to follow these standards unless he wishes to make an installation which can be certified just as heating installations are certified by the *Chicago Master Steam Fitters Association*. Brief experience has demonstrated that many owners refuse to accept plants which cannot be certified as complying with these standards and it is believed that their use will be effective in improving the general quality of the work done without increasing the cost of quality results.

In many cases mistakes in computation will be discovered and corrected before it is too late, as a result of expert checking of data by an independent qualified engineer, just as countless corrections in boiler and radiator and piping selection have been discovered and corrected under Certified Heating practice. Anyone may have his plans and specifications certified for a small fee and the work is done by a regular salaried employee of the two contracting organizations. In case corrections are required no information is disclosed except to the person or firm by whom the project is submitted. The central examining staff will gladly explain the standards, but cannot design apparatus, recommend any specific type of design or system nor suggest any particular manufacturer.

* Consulting Engr.

Since this paper was presented, the Chicago Committee on Air Conditioning Standards representing a cross section of the entire air conditioning industry has been organized and has completed a set of standards in somewhat different form. In these standards the results to be attained are emphasized. The methods of attaining these results are limited to suggestions embodied in explanatory notes.

It is probable that these new standards, not yet available for publication, will supersede the standards mentioned in this paper as the basis for Certified Air Conditioning. Mr. John Howatt, President of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, is Chairman of this new committee and Samuel R. Lewis is Secretary.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June, 1935.

If a design is certified, the combined resources of the two contracting organizations assure proper installation and operation of the plant because through their influence with the contractor supplemented by the experience and ability of the governing committees, bad workmanship will be corrected or even apparatus of manifestly improper size will be replaced.

The standards for certified air conditioning for comfort for the Chicago district were developed cooperatively because of the urgent demand by the public for some measure of protection against incompetent designs.

These standards, like any new and radical departures, were developed from the experiences of the various participants in their creation, but were scrutinized in all details with much the same care as might be given to a new municipal building code with public hearings on every clause. In this case the committee, after making an assumption, consulted with the most distinguished available authorities among engineers, contractors, architects, and manufacturers. In practically every case the standards were modified reasonably to conform with the suggestions of every communicant.

One of the interesting psychological difficulties was to persuade people to read the standards and then having read them to make them understand that these rules simply set up limits of fair practice.

As an example of the value and desirability of having air conditioning standards it is merely necessary to cite the case of one very large installation involving hundreds of tons of refrigeration on which bids and designs were requested from various manufacturers and contractors. The bidders were furnished with floor plans of an existing building and with nothing else, but were expected to submit complete firm proposals. In this case no one knew what outside air conditions or inside air conditions should be assumed, what condensing water temperatures nor what type of compressor or refrigerant should be used.

The common experience of Chicago building owners, who heed the request of manufacturers' representatives for permission to bid on proposed air conditioning projects, has been that the proposals varied to such an extent that no confidence could be placed in the bids. Several very interesting projects for which funds were available, were abandoned, because the owner said "if air conditioning is as crazy as these proposals indicate with as much as 100 per cent difference on tonnage for the same job and if all bidders are anxious to sell me cast-iron, copper-plated guarantees, I'll just wait until I become convinced that the bugs have been driven out and you fellows learn what you are about."

Confronted with this state of affairs, the heating contractors organization approached the Consulting Engineers Division of the *Western Society of Engineers* upon the subject with the result that arrangements were made for official appointment of representatives by various interested organizations. Of a committee of seven men, five hold membership in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. At the time of organization the A.S.H.V.E. had not appointed its committee on Standards for Air Conditioning.

The first action of the Committee was to prepare a definition of air conditioning, which closely resembles the official definition adopted in 1932 by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS as a part of their Ventilation Standards.

Because of the many existing definitions of the term *air conditioning* and

the difficulties involved in getting universal acceptance, it is evident that engineers are not of one mind on the subject.

The Chicago standards require that if an installation of Certified Air Conditioning is to be made the plans and specifications must show how the load demand was developed. Here is the clause which covers this point:

*2. INFORMATION FOR CHECKING. The plan and/or specifications of all air conditioning installations submitted under this standard shall set forth the heat transmission factors for barriers and the amounts and sources of heat loss and gain. The design temperatures and relative humidity shall be recorded for air outside and inside, also for condensing air or water, and for the heat transferring medium.

These governing design factors and temperatures shall be such as to produce an effective result at least equal to those set forth in these Standards.

This clause means about the same as a requirement that the size of a boiler for a heating system must be stated:

The clause about guarantees is as follows:

3. GUARANTEES. Guarantees of performance in specifications on which competitive bids are taken where definite minimum sizes of apparatus are specified, shall be limited to the requirement that each piece of apparatus shall deliver the rated output stipulated by its manufacturer, and that the contractor's performance guarantee shall include only the adjustment and coordination of these various capacities.

This item is an ancient bone of contention. No self-respecting consulting engineer will include in his specifications the all-embracing guarantee that the contractor must furnish everything that is omitted from the specifications which the engineer or owner might think should be furnished in order to make a complete and perfect job. An engineer is reluctant to require that the contractor, competing in price with others who perhaps are not competent or honest, shall guarantee that the sizes of apparatus which the contractor didn't select shall be big enough to do everything that every one thinks ought to be done.

The committee may modify this clause to the extent that if the owner wishes to receive from the contractor a guarantee of sufficiency and of operating performance the contractor shall be required to check the capacities of equipment and may increase such capacities at his own option, stating the difference in price if any, without prejudice or sacrifice of good will, so that all may enter into a guarantee of performance with their eyes open.

It is realized that an owner may fairly require the added insurance of a bonded contractor and may require that large sums of money shall be withheld pending fulfillment of a guarantee, provided that all concerned understand the situation.

The requirements of Item 3 require compliance with the current A.S.H.V.E. GUIDE for design coefficients of heat transfer through barriers and for heat from appliances, people, etc.

Design temperatures and relative humidities are specified as follows in Items 5, 6, 7, and 8:

Outside in winter..... — 10 F
Inside in winter.....70 F and capable of producing 30 per cent relative humidity when the outside condition is 30 F and 40 per cent relative humidity, with explanations about danger of frosting.
Outside in summer.....95 F and 40 per cent relative humidity
Inside in summer.....80 F and 55 per cent relative humidity

* Sections 2, 3, 10, 11, 12, 13, 14, 15, 16, 17, 18, and 19 quoted by permission from the pamphlet, Standards for Certified Air Conditioning for Comfort issued by *Chicago Master Steam Fitters' Association.*

It is probable that a clause will be added to make clear the reasonable understanding that any Effective Temperature equivalent to 80 F and 55 per cent would be acceptable.

In Item 9 a small allowance for infiltration is provided for summer even though some people argue that infiltration in summer is ordinarily negligible. The allowance is stipulated because of the belief that when room-located unit recirculating cooling machines are used there must be local provision for condensing the moisture of inleaking air.

The stipulated allowances for infiltration are 60 cu ft per hour per square foot of all the area of windows and doors in outside barriers in winter. This is a simple approximation and does not in any way prevent the designer from computing crack width and wind velocity nor militate against his prayer that the crack will not increase or decrease nor that the wind will not blow more belligerently than at the usual 15 mile per hour rate.

The infiltration allowance when cooling is suggested as a minimum of 15 cu ft of air per hour per square foot for all of the area of these same openings and explanation is made that the sensible and latent heat for this air must be figured out and added into the load.

Minimum solar heat allowances are set forth in a very simple table as follows:

10. DESIGN ALLOWANCES FOR HEAT GAIN DUE TO SUNLIGHT. Shall not be less for each exposure than those shown by the accompanying table:

	N	NE	E	SE	S	SW	W	NW
		7 A.M.	8 A.M.	9 A.M.	12:00	3 P.M.	4 P.M.	5 P.M.
Window	0	100	130	110	70	110	130	100
Skylight	0	60	100	135	180	135	100	60
Wall	0	0	2	2	2	3	3	3
Roof	0	0	0	2	4	5	5	4

The above allowances per square foot of sun-exposed area for walls and roofs are on a basic heat-transfer resistance of U-0.3 and a gray surface color. Variations in the value of U and in the color co-efficient ordinarily affect the result to a negligible extent.

(a) For restaurants and similar rooms in which the heat gain, due to occupancy and due to process work, is an important factor, the total heat including sunlight heat should be computed as of the hour when all heat other than sunshine is at the minimum.

The total heat including sunlight heat and the heat other than that from sunshine should be computed also at the hour when sunshine is at the maximum. Whichever of these two heat inputs is greater should be used.

This explanation is made to avoid possible misinterpretation and the provision of unnecessary capacity such as might be called for if the maximum heat during the 12:00 noon hour of a restaurant having a large West sunshine exposure were added to the 3 p.m. sunlight heat.

(b) When computing the refrigerating capacity for any building having many rooms and several different sunshine exposures, it will be necessary to re-compute the sunshine heat effect for the building as a whole and to use only the greatest time-direction factor for the building as a whole, in determining the tonnage. Otherwise,

for instance, the total East sunshine heat for many rooms might be added to the total West sunshine effect for many rooms, whereas only the greater of these two sums need be provided.

(c) When computing the sunshine heat effect for any room which has more than one wall exposed to direct sunlight at any given time, only the heat input for the side having the greatest value is to be considered.

When this scheme of allowing for solar heat is worked out in practice its simplicity is appealing. There is nothing to prevent the designer from making a smaller arbitrary guess as to how much smoke there may be to reduce the sunshine intensity or as to how much faith he may place in a white wall remaining white—or in someone pulling down the awnings. The values stated are minimum values and they may well be increased. All is by no means yet known as to the lore of proper solar heat allowances.

To this end these standards have the following to say about allowance for shade from trees, buildings, and the like:

11. DESIGN ALLOWANCE FOR SHADING FROM SUNSHINE shall be made only when the plans and specifications specifically show such allowance, and give assurance that the owner is cognizant of the reduction in capacity on this account. A design reduction on account of shading, which may be temporary if the shading structure should be removed, or which depends upon human operation, is of questionable wisdom. If the shading is present, of course the operating expense will be reduced.

Item 12 covers design air quantity. The present requirements of the ventilation code of the city of Chicago for school rooms and auditoriums, factories, and similar occupied space having fixed seats, is a minimum of 1500 cu ft per hour taken from outside for each sitting. If 12 sq ft of floor is allowed per person this means approximately, 2 cu ft of air per square foot of floor per minute.

For stores this quantity is reduced to 1.8 cfm per square foot of floor for basements and to 1.5 cfm per square foot of floor for first stories.

For restaurants 1.65 cfm per square foot of floor holds, except that in kitchens, where double this amount, 3.33 cfm per square foot of floor is required or a change every 3 min must be provided on the basis of a 10 ft ceiling.

There is some tendency toward unofficial leniency for plants which are capable of handling code requirements when heating or when cooling without refrigeration, if during the use of refrigeration they do not recirculate more than 66⅔ of the air. This is a reasonable attitude and is said to be in effect pending adoption of the new Chicago Building Code which probably will legalize two-thirds recirculation.

This is in explanation of Items 12 and 13 of the Standards, which read as follows:

12. DESIGN AIR QUANTITY. (a) For winter air conditioning the design air quantity shall be not less than the requirements of the City of Chicago for ventilated rooms.

In rooms which do not come under the Code as of February, 1935, such as residences, a minimum air circulation of 4 changes per hour shall be provided. In such rooms where the occupancy by people is the governing factor, an air circulation of not less than 25 cu ft per minute per person shall be provided.

(b) For summer air conditioning the design air quantity shall be sufficient to extract the surplus heat at the air inlet and outlet temperatures specified, and in addition shall be not less than the requirements of the City of Chicago, for ventilated rooms.

In rooms which do not come under the Code as of February, 1935, such as residences, a minimum air circulation of 4 changes per hour shall be provided. In such rooms where the occupancy by people is the governing factor an air circulation of not less than 25 cu ft per minute per person shall be provided.

13. RECIRCULATION AND AIR FROM OUTSIDE. Provision may be made in the duct design for recirculation of not more than 66⅔ per cent of the circulated air, with 33⅓ per cent air from outside, which always shall be provided to dilute odors during occupancy. The inlets and ducts shall have sufficient capacity to permit the introduction of 100 per cent of air from outside for operation during the periods of intermediate outside temperature.

There is no particular authority for the 5 deg temperature difference suggested in sub head (a) of Item 14, except that it appeared wise to state some value. The 5 deg is a matter of judgment. In sub head (b) the rule of 2 deg per foot in height above the floor is based on the experience of the group and would allow a 20 deg temperature spread in an 11 ft ceiling provided that the inlet grille is not more than 12 in. high. Surely this seems a liberal allowance. The minimum of 100 linear feet per minute air velocity for horizontally discharging openings probably is far too low. In the judgment of some a minimum of 300 linear feet per minute would be more nearly correct. Item 14 is as follows:

14. DIFFUSION OF ENTERING AIR. The location, shape, physical characteristics of, and air velocities and temperatures through the inlets and outlets for conditioned air to the rooms shall be such as will promote distribution of the circulation throughout all occupied portions without causing discomfort.

(a) Downward discharging air inlets in rooms shall be provided with safeguards to prevent drafts at the occupied zone. These safeguards shall include plaques, injector type grilles, or other devices which shall discharge the air at an angle to the vertical and which will prevent air colder than the room air from falling vertically downward at speeds capable of causing discomfort. Unless such safeguards shall be provided, the design entering air temperature from inlets discharging vertically downward shall be not more than 5 deg colder than the room air at the approximate level of the inlet.

(b) The difference in temperature between the entering air and the room air for horizontally discharging supply openings in side walls with conventional registers when cooling shall never exceed 2 deg per foot in height from the floor to the bottom of the opening.

Horizontally discharging air inlets in rooms when placed higher than head level shall be designed for sufficient velocity of discharge to bring about diffusion without drafts, especially when the entering air is cooler than the room air. The minimum design air inlet velocity shall be 100 linear feet per minute. The entering air velocity may be rapid for such openings, provided that high speed jets of air do not rebound from columns, walls, etc., so as to create discomfort to people.

(c) Upward discharging air inlets in rooms shall be designed with arrangements, especially when delivering air cooler than the air in the room, so that a velocity of at least 300 linear feet per minute may be obtained so as to promote diffusion.

In Item 15 a stipulation is made that the refrigerant and the refrigerating apparatus shall comply with the rules of the Board of Health and of the *Chicago Master Steam Fitters.* These rules in a general way are no more severe than the A.S.A. Safety Code covering the installation of refrigerating apparatus sponsored by the *American Society of Refrigerating Engineers.* There are rather drastic speed limitations given for reciprocating compressors, based on the experience of the contractors organization but checked with several prominent manufacturers. The speed limits were made rather low because under such circumstances the machines assuredly should have a long life and

will have a desirable surplus capacity as a reserve against mistakes in estimating the demand or against possible future increases in load.

The condenser areas specified are also liberal, to provide against scale and dirt and to give a reasonable reserve capacity. The condensing water temperatures are standards and their inclusion gives a basis for reference and safeguards fair competition.

It should be explained that the specific limitations of Item 15 will be administered liberally, and that if any manufacturer should evolve some especially efficient process or device which contravenes them the Committee which is continuing will consider proper revisions.

15. REFRIGERATING APPARATUS AND REFRIGERANTS. All refrigerating and cooling equipment, piping and specialties used in connection with air conditioning shall be designed and installed in accordance with the requirements of the Chicago City Department of Boiler Inspection and Cooling Plants, the rules of the Chicago Board of Health, and the standards of the *Chicago Master Steam Fitters' Association.*

The following are minimum acceptable capacities for the various parts of the refrigerating apparatus to be used in comfort cooling:

(a) Evaporator; pressure difference between inlet and outlet not to exceed one pound per square inch.

(b) Compressor; the load permitted on reciprocating refrigerating compressors under these standards shall not exceed the output delivered at the speeds set forth below; and the maximum operating speed shall be limited accordingly.

For Multicylinder machines delivering not to exceed 25,000 Btu per hour per cylinder	400 rpm
For similar machines delivering more than 25,000 and less than 50,000 Btu per hour per cylinder	360 rpm
For similar machines delivering 50,000 and less than 100,000 Btu per hour per cylinder	290 rpm
For similar machines delivering 100,000 and less than 200,000 Btu per hour per cylinder	220 rpm
For similar machines delivering 200,000 and less than 300,000 Btu per hour per cylinder	175 rpm
For similar machines delivering more than 300,000 Btu per hour per cylinder	150 rpm

It is understoood that reasonable variations may be interpolated between these limiting relations as they have been taken from a plotted curve.

(c) Condenser, normal condensing pressure shall correspond to 86 F for the refrigerant used. For each 12,000 Btu per hour heat transfer, the effective water cooling surface shall be not less than the following:

Double pipe type, made of copper	6 sq ft
Shell and coil type with copper coil	8 sq ft
Shell and tube type with steel tubes and equipped with swirling devices	8 sq ft

If other material than copper is employed correction shall be made for any difference in conductivity.

(d) Condensing water; not cooler for Chicago city water than 76 F. Not cooler for cooling tower than 80 F.

(e) Power required from the driver in no case shall exceed the manufacturer's normal rating.

(f) The receiver shall be of sufficient capacity to permit pumping out the evaporator and the liquid lines without increasing the condensing pressure more than 50 per cent above the normal pressure (corresponding to 86 F) for the kind of refrigerant used. The refrigerant should be pumped out of the evaporators and stored in the receiver during the period when the apparatus functions for heating or is exposed to room temperature of 70 or 80 F.

(g) Apparatus such as evaporators, condensing units, expansion valves, etc., shall be installed with sufficient line valves and proper fittings, to permit efficient servicing, or quick replacement of parts without exposing the interior of the refrigerant-con-

taining parts of the system to air and moisture. Refrigerant lines shall be of proper size to prevent any appreciable pressure difference between the evaporators and the compressor.

(h) Accessibility for servicing and repair of all parts of air conditioning systems must be provided.

Item 16 deals with gages and weights and bracing of ducts and housings and reflects the experience of sheet metal contractors especially as to the calls which they have had for repairs and maintenance. Too many existing duct systems have their bottoms bulging down and leaking due not only to the periodical entry by men for cleaning purposes but also due to workmen entering during the process of erection. Some standard of this kind has been needed for a long time, though approximately the same data have been available. Item 16 is as follows:

16. METALS USED FOR HOUSINGS, TANKS, DUCTS, ETC., in air conditioning installations shall be of moisture resisting character and of such weight and with such bracing as will prevent vibration. The ducts shall be substantially air tight.

Each branch of a supply duct system shall have facilities for adjusting the air volume delivered. Preferably these dampers shall be at junction points and alternatively they may be at the registers.

If galvanized steel sheets are used they shall be not lighter than the U. S. gauges listed below with structural steel exterior bracing as shown.

SERVICE	GAUGE OF METAL	SIZE OF BRACING
Intakes	20	
Heater casings	16	
Filter, heater and other chambers	20	1¼ in. x 1¼ in. x ⅛ in. minimum
RECTANGULAR DUCTS:		
Less than 18 in. wide or high	26	
18 in. to 29 in. wide or high	24	
30 in. to 47 in. wide or high	22	1 in. x 1 in. x ⅛ in.
48 in. to 60 in. wide or high	20	1¼ in. x 1¼ in. x ⅛ in.
Over 60 in.	18	1½ in. x 1½ in. x 3/16 in.

If the duct has width or height of more than 24 in., there shall be transverse exterior braces for such surfaces not more than 48 in. on centers. Where "S" cleats and drive cleats are used to connect ducts, formed angle cleats may be used in lieu of the stiffener angles. The same gauges as specified above shall apply, for circular ducts of diameter corresponding to the given widths.

Some revision of Item 17 should be made since the clause concerning air filters should properly be in a separate item and should be elaborated. This new item might mention the desirability of a fireproof type of air filter. The uninitiated might be surprised to learn how often air washers have been installed

without adequate waterproof safing and have thereby caused great expense and embarrassment.

17. HEAT TRANSFER APPARATUS. All convectors where condensation might occur when cooling shall have drain pans with tubing running to drip with an air break above a trapped sewer connection or its equivalent.

No fixed spray humidifier or spray dehumidifier shall be installed above the basement of a building unless there shall have been installed under it a water-tight safing graded to a drain, so that in case of leakage or overflow there shall be no danger of water damage to the rooms under the apparatus.

The air velocity through the air-way of a spray humidifier or spray dehumidifier shall not exceed 500 linear feet per minute.

Eliminator plates shall be provided for spray humidifiers, for spray dehumidifiers and for evaporator-convectors and shall be made of some especially durable material so designed as easily to be replaced.

Air filters always shall be easily accessible for inspection and removal without danger of spilling dust to leeward of themselves when the substitution is made.

Item 18 deals with objectionable noise from air conditioning apparatus and merely emphasizes the suggestions made in our proceedings during many years that the floating of the foundation which carries the apparatus is more effective than merely to attempt to float the apparatus itself especially if there is a belt drive and necessity for anchoring rigidly such a thing as a small electric motor. The speed limitations for air velocity in ducts may be criticized as ultra conservative, but again the defense is that these standards cannot take into consideration proprietary devices. Assurance is given that a liberal interpretation by the plan examiner and the Chicago committee will be taken where the effectiveness of the proprietary devices is assured.

18. NOISE CONTROL. All moving apparatus used in air conditioning shall be fastened securely to its own foundations, which should be a single unit for both driver and driven apparatus, and this foundation shall be carried on a noise-inert substance such as rubber in shear, or cork under and around the edges of the foundation, without any direct contact between the foundation and the building structure.

No metal duct shall connect directly to any fan or housing without a sound-inert junction, such as of canvas.

Piping connections from pumps where practicable shall be made with rubber hose, with the machinery on a floating foundation.

Sharp edges of metal facing an air current shall be rounded or streamlined rather than left sharp and rough. Partitions, housings, and large ducts must be so heavy and so well braced that there will be no vibrating or rattling when air is being delivered.

Intermediate deflecting vanes to prevent building up undue pressure and velocity inside the outer curve of trunk line elbows are desirable.

On account of noise danger the following air velocities shall not be exceeded in those parts of the ducts of air conditioning systems closer than about 50 feet from the room outlets.

Fan outlet	1500 lin ft per min
Trunk duct	1200 lin ft per min
Through free area of convectors	1000 lin ft per min

Where fans, motors, compressors, and the like are placed in rooms under or otherwise adjacent to occupied rooms, provision shall always be made to reduce noise transmission. Suggested means are double walls, double ceilings, sound absorbent material and the like.

Item 19 was called forth by experience with many plans and specifications which were hopelessly incompetent as to insulation. This item should be useful

when calling attention to such discrepancies and may fulfill the need of some standard which might be cited as an example.

19. INSULATION. Insulation on pipes and air-ways when heating, is designed to reduce transmission to the surrounding air, and is under no danger of becoming moist except in case of a liquid leak.

Insulation on pipes and air-ways when cooling, in addition to reducing heat transfer from the surrounding air to the cooler inside, must prevent the surrounding air from reaching its dew point and thus depositing moisture. The dew point temperatures for conditions of the air around a cool duct or pipe may be read from a psychrometric chart, but are suggested as follows:

TEMPERATURE OF COOL SURFACE	AIR TOUCHING THE COOL SURFACE					
	DbF	Rh at DP	DbF	Rh at DP	DbF	Rh at DP
40	60	48%	70	34%	80	25%
45	60	58%	70	40%	80	29%
50	60	70%	70	50%	80	35%
55	60	85%	70	60%	80	42%
60	60	100%	70	70%	80	50%

This indicates that if a cold surface at 40 F is permitted to cool the air which comes in contact with its outer surface, to 60 F and 48 per cent Rh there will be a deposit of moisture akin to that which one sees in summer beading the surfaces of uninsulated cold water pipes in basements.

In air conditioning in which refrigeration is used, therefore, all chilled surfaces must be insulated not only as a measure of control of heat waste, but also to prevent damage due to water drip.

For ducts carrying air not colder than 40 F, a ½ in. layer of board or quilt insulation will serve, provided that the chilled areas are all covered so that air does not pass against the cold surface. For housings around evaporators and for spray dehumidifier housings ½ in. of quilt insulation and ½ in. of board insulation with alternating joints and thorough sealing, have given satisfactory service. Cork 1 in. thick properly sealed, gives satisfactory results, especially for spray dehumidifier housings which carry direct expansion evaporators also.

For low side refrigerant piping cork insulation is acceptable, of thickness commensurate with the temperature difference.

For systems which use cold water piped to the convectors for summer only, any anti-sweat pipe insulation 1 in. thick is acceptable. Where the same pipes convey hot water in winter and cold water in summer, a combination insulation shall be used, which will withstand both extremes. Compressed mineral wool has this faculty, though there is evidence that any good quality steam pipe insulation will serve satisfactorily provided that there are no exposed chilled surfaces to condense the damaging dew.

The standards end with pertinent extracts from the city codes covering local requirements as to permits and fees and quotes local rules as to certain safety features. The situation concerning evaporators for chemical types of refrigerant placed in air ducts of ventilating systems is not entirely satisfactory and undoubtedly must be clarified soon.

There are many large factories having work rooms crowded with employees with inadequate exits to the outer air which are cooled by direct expansion

ammonia, sometimes in convectors and frequently with direct expansion coils in air washers where they are sprayed with water. Some experienced refrigerating engineers defend this practice. Others equally experienced condemn it. The Chicago building code prohibits such practice, and the Standards for Certified Air Conditioning rest on the Code.

There is need for clarification of this matter and research may be required because there is such a wide variation in the properties of the refrigerants now in common use, and in the pressures and in the kinds of metal and in the safety devices.

Many engineers fear that some serious accident may happen before proper restrictions and safeguards shall have been developed and enforced. A suggestion is made that this particular matter properly might be pursued further by a joint Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers.*

The Committee which developed these standards for Chicago realizes that improvements can be made and hopes to keep them from becoming too detailed and too voluminous. Changes can be made from time to time as experience in administering them dictates.

DISCUSSION

S. H. DOWNS: I am thoroughly in accord with the purpose of these standards and believe that it will be helpful in placing the business on a definite foundation so that purchasers and users of the systems can believe that all of us know something about what we are doing.

In order to make these standards of the most use and for them to receive the respect that is due them, I believe that it is necessary to review carefully the specific limitations that are put in the standards to see that they agree with up-to-date practices and allow the necessary means for adoption of new methods. If these design limitations are not sufficiently broad in scope, it will simply mean that these standards will become obsolete in a short time.

In Section 15a, it will be noted that the difference in pressure between the inlet and outlet of evaporators is specified not to exceed one pound per square inch. This value is not in accord with present day practice as recommended by the coil manufacturers for direct expansion cooling coils. There is nothing inherently wrong in having a somewhat higher pressure drop through the coil if this seems desirable, and in order to place this limit at a more practical value, I would suggest changing it to 3 lb per square inch. This value would prevent the use of excessively high pressure drops through the evaporator coil and at the same time would permit flexibility in design to meet varying conditions as we find them.

Part b in this same section shows a table of maximum operating speeds for reciprocating refrigerating compressors, and I believe that these values are all too low to permit the use of various standardized designs that are now on the market and which have proved their worth. It does not seem from these values that the various manufacturers of refrigerating compressors have been consulted to see whether or not the various machines could meet these specifications according to the present practice of design.

In Section 18, Noise Control, there is a statement that the supports under the driver and driven apparatus shall all be in a single unit with the noise-inert substance below this single piece. This description covers one type of noise isolating base, but there is another type of base which performs the opposite principal, in that the driver and driven apparatus are purposely separated, and the noise-inert substance is below each one separately, and that in turn mounted on a single base. In the case of a belt driven centrifugal fan, it is a very distinct advantage to have the

motor separated from the fan by the noise-inert substance so that the motor noises cannot be transmitted through the fan proper and on into the air stream.

Also Section 18 should include some warning about taking care of the acoustical properties of the conditioned spaces themselves, as a great many buildings are built so that no matter how quiet the air conditioning equipment may be, the building itself will magnify the noises. The way the section is now written, considerable warning is given about eliminating noise in the air conditioning system itself, but nothing is mentioned at all about taking care of the bad acoustical properties which are often found in the conditioned spaces.

The latter part of Section 18 on Noise Control specifies certain velocities that shall not be exceeded closer than 50 ft from the room outlets. I presume that this distance of 50 ft is measured along the path of the air and not the nearest line between the two locations in question. I suppose that this table of velocities is merely an effort to provide some definite values that are felt to be based on the noise standpoint, but outlet velocities at the fan and in the air ducts are not necessarily a measure of noise, and it is entirely possible that these velocities may be met and other difficulties might cause the system to be noisy. Also, it is entirely feasible to use higher velocities in all these locations provided the necessary noise deadening means is installed in the air passages. On some jobs it may be absolutely necessary due to space conditions, to choose higher fan outlet velocities and duct velocities, and then absorb the noises with sound deadening means in the ducts, although the standards as written apparently would not permit this. Furthermore, the noise emanating from the fan is not dependent only on the outlet velocity of the air, but is also dependent upon how high static pressure the fan is working against, and it also varies with different types of fans. The whole noise situation is really a very complex problem and I do not believe that it can be really covered with such a limited set of conditions as outlined by Section 18.

J. H. VAN ALSBURG: In connection with Section 18 I would like to mention that I am not entirely in accord with the maximum velocity set up to eliminate noise. If the maximum velocity 50 ft from the outlet is 1200 fpm and the expansion is in the approach, the outlet velocity will often be too low for proper air distribution. Many installations are operating satisfactorily where the outlet velocity is very much in excess of 1200 fpm and I think an allowance should be made in these standards for such conditions.

D. W. NELSON: In Section 18 on Noise Control, I expected to find limitations on the tip speed of fans.

W. A. DANIELSON: I have been very much interested in this paper and particularly the discussion that has taken place. My membership in the American Institute of Electrical Engineers which has extended over a period of some 30 years, and my work in all branches of engineering, has led me to the conclusion that standardization should be carefully done. This is particularly true of a young science and air conditioning in its general application is quite young. We have been trying to write specifications for the ordinary small heating boiler for many years and yet we have a committee now working on these standard specifications.

There is a philosophy behind this whole question that we should not overlook. Law and codes will not take the place of a reputation. It has been my duty to employ many engineers and in determining what an engineer can do I have investigated what he has done. In other words, I have gone back to his reputation. This was not made by law or code but by his own inherent ability. Let us not make the mistake of thinking that we can lay down a code that will eliminate the difficulties that come through ignorance. Let us rather try to convince the people that want air conditioning that they must go to successful men to get a satisfactory job. If we try to do otherwise we shall keep costs high and defeat the general use of air conditioning more than a few *sour* jobs will do. The thought I am advancing is not new. All industry has been built on reputations and the elimination of those who do not know by the failures that result from their work.

In Memoriam

Names	Joined the Society	Died
Charles W. Christian	1913	Jan. 1935
Ralph Collamore	1904	Nov. 1935
George W. Duncan, Jr.	1923	June 1935
W. Roy Eichberg	1929	July 1935
George M. Getschow	1906	Nov. 1935
Helen R. Innis	1918	Jan. 1935
Stewart A. Jellett	Charter Member	Apr. 1935
Charles D. Kirk	1909	Nov. 1935
John M. Robb	1913	May 1935
John D. Small	1910	July 1935
Earl J. Vermere	1929	Dec. 1935

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